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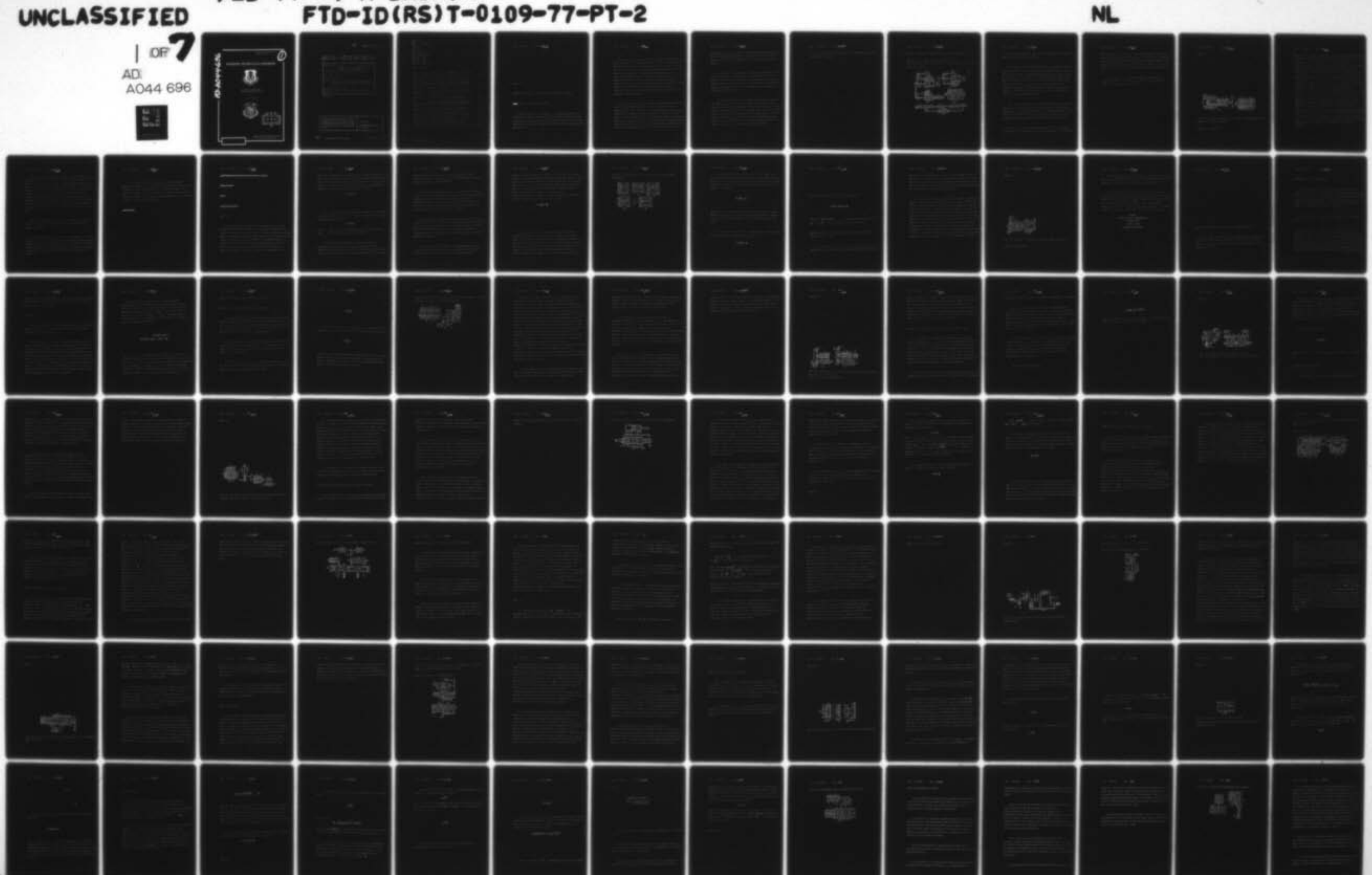

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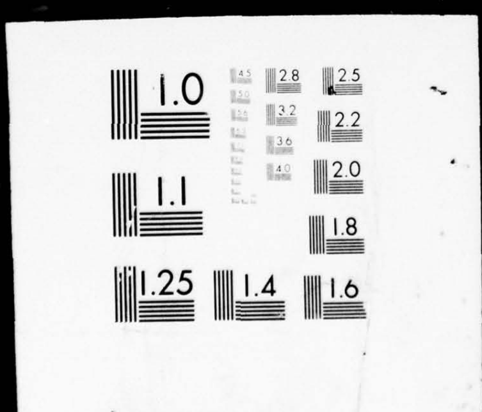

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FOREIGN TECHNOLOGY DIVISION

AIR FORCE SYSTEMS COMMAND

HYDRAULIC DRIVE AND
HYDROPNEUMONAUTOMATION

by

T. M. Bashta

FOREIGN TECHNOLOGY DIVISION

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HYDRAULIC DRIVE AND HYDROPNEUMOAUTOMATION

By: T. M. Bashta

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Table of Contents

Page 159.

CHAPTER IV. Hydraulic servomechanisms (hydraulic boosters).

[redacted]/ELEMENTS OF HYDRAULIC BOOSTERS.

Hydraulic booster - the totality of hydraulic apparatuses and volumetric hydraulic engines, in which the motion of control device is converted into the motion of the controlled member of large power, matched with the motion of control device along speed, direction and displacement.

Slave/servo type hydraulic booster is power hydraulic drive in which actuating mechanism (output/yield) reproduces (it tracks) the law of the motion of the control device (inlet), for which in system is provided for continuous communication/connection between exit and input computers, which is called feedback. Otherwise by the slave/servo hydraulic drive is understood hydraulic drive with the automatic control in which the speed of motion or the course (rotation) of the driven/known component/link of volumetric hydraulic engine changes according to the determined law depending on previously not the known variable, which affects from without the volumetric hydraulic drive.

Names of this drive - the "slave/servo hydraulic booster", and also the "slave/servo hydraulic drive" are based fact that the output/yield of this hydraulic booster automatically removes through feedback the appearing disagreement/mismatch between the control pressure (input signal) and the reciprocal action (output signal). By inlet either input signal, here is understood any intentional effect (displacement, the speed and the other parameters), communicated to the sensor of disagreement/mismatch from setting device, while by

output/yield or output signal is understood the effect, developed with actuating element (hydraulic engine) with the aid of which is realized the required displacement of the given operating unit of machine.

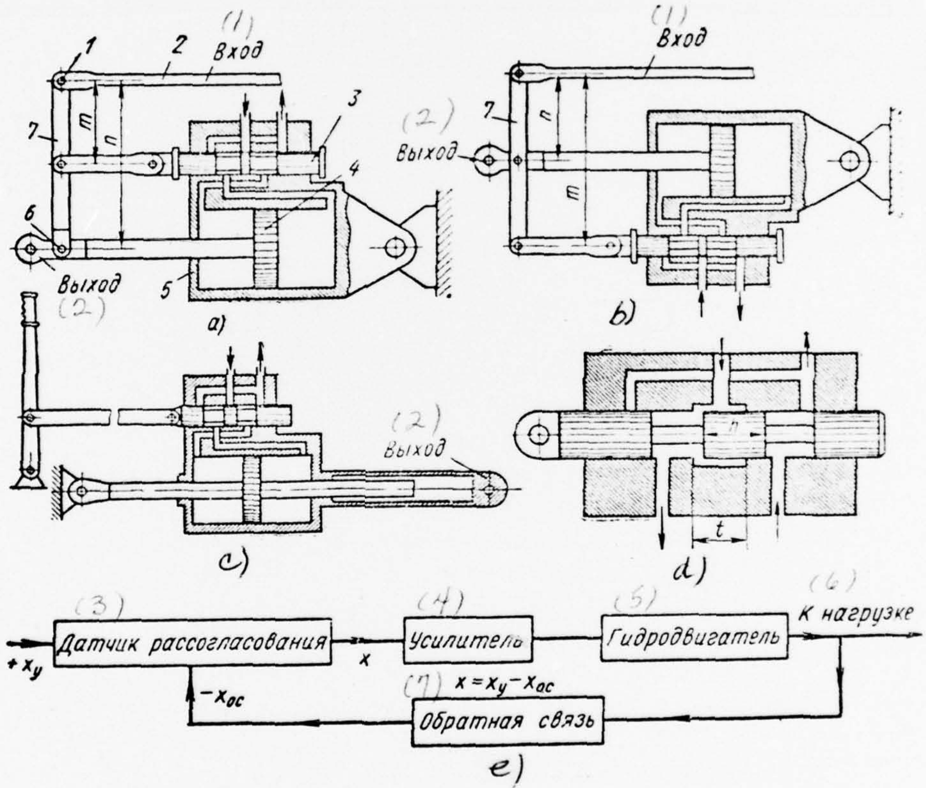
The hydraulic servomechanisms had extensive application in the different branches of technology and in particular in the control systems of contemporary cargo vehicles, including motor vehicles, marine vessels, aircraft and other flight vehicles. The especially widely servomechanisms are used for copying the form of workpieces and copying displacements.

The designation/purpose of the slave/servo hydraulic drive of the control system lies in the fact that, in order to move load (loaded operating unit) according to the assigned law and at given speed, providing in this case the required intensification of output power, obtained by using an energy of the applied liquid. By load here is understood the complex of the static and dynamic forces, which act at the output/yield of actuating mechanism (cylinder of hydraulic motor or the stock/rod of actuating cylinder).

The degree of the intensification of output power (power gain) is virtually not limited.

Fig. 115. Diagrams of slave/servo type hydraulic boosters.

Key: (1). Inlet. (2). Output/yield. (3). Sensor of disagreement/mismatch. (4). Amplifier. (5). Hydraulic engine. (6). To load. (7). Feedback.



At high output power the power of input signal can be decreased down to negligibly low value (0.5 W).

For a tracking in the hydraulic boosters in question is commonly used the negative feedback, which transmits output signal to inlet (to control device). This communication/connection is called negative as a result of the fact that the effect, which enters from it to the inlet of hydraulic booster, is opposite on sign to the basic input effect (control knob, etc.). When the negative output signal, transmitted through this feedback, becomes equal to positive input signal, the hyiraulic engine of amplifier is disconnect/turned off (ceases the power supply of engine).

There are different forms of the feedback from which in systems and devices of automatic control, in particular in electrohydraulic servomechanisms, most widely is applied the follow-up direct feedback of output/yieli with inlet on position, which possesses the high accuracy/precision of tracking and stability against vibrations.

By the design concept of the last/latter form of feedback, are distinguished the hydraulic boosters with lever/crank feedback (Fig.

115a and b), the coefficient of feedback (amplification factor) of which it is possible to change with the selection of the relationship of arms m and n of differential lever 7, and hydraulic boosters with rigid single feedback (Fig. 115c and 116a), in which sensing device (valve, nozzle, etc.) it is placed directly on exit component/link.

The diagram of the slave/servo hydraulic booster with the hydraulic engine of rectilinear motion and by rigid lever/crank feedback is represented in Fig. 115a.

Page 161.

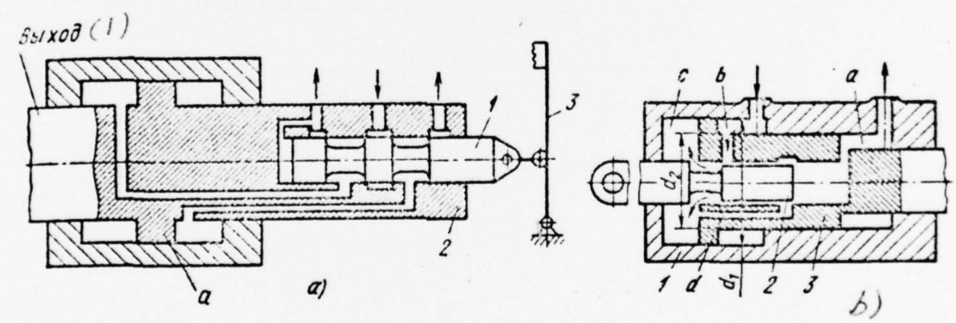


Fig. 116. Hydraulic boosters with the valves, arrange/located in the piston of actuating cylinder.

Key: (1). Output/yield.

This hydraulic booster consists in essence of the same cell/elements, as hydraulic drive examined above, differing from them only in terms of the presence of follow-up direct feedback in the form of differential lever 7, that covers distributor (inlet) 3 and hydraulic engine (output/yield) 4 (see Fig. 115e), and also fact that for providing equal conditions during piston stroke into both sides are applied cylinders with bilateral stock/rod (see Fig. 26b). In moving thrust/rod 2, connected with control knob, is moved the point of 1 differential lever 7 feedback with which are connected the stock/rods of actuating cylinder 5 and of the valve of distributor 3. Since the forces, which counteract to the displacement of the valve of distributor, considerably the less appropriate forces, which act into the system of power piston 4, point 6 can be considered in the beginning of the motion of thrust/rod 2 as motionless, in view of which its motion will cause through lever 7 displacement of the plunger of the valve of distributor 3. As a result during the displacement of valve from free position to the value, greater h-t/2 (see Fig. 115d), liquid will enter the appropriate cavity of cylinder 5, which will cause the displacement of piston 4 and, consequently, also the points of 6 output/yield to certain way, proportional to the displacement of thrust/rod 2 (to deflection of the hanile of control).

After the motion of thrust/rod 2 will be ended, continuous to move piston 4 will report through the lever to 7 feedback to the plunger of the valve of distributor 3 displacement, opposite to that which it obtained thus far during the displacement of thrust/rod 2 controls. Since in this case expenditure slide valve ports will be as a result of the reverse motion of its plunger gradually to be covered, amount of liquid, which enters cylinder 5, decreases, in consequence of which the speed of its piston it will decrease until the plunger of valve arrives to the position in which the windows completely will overlap, in this case speed it will become equal to zero.

During the displacement of the plunger of valve to opposite side, the motion of all cell/elements of adjuster will occur in opposite direction.

In actuality, of the various stages of the motion of inlet and output/yield of the servomechanism in question with follow-up direct feedback there does not exist, and both motions flow/last virtually simultaneously, i.e., it occurs not stepped a continuous "tracking" as actuating mechanism (output/yield) after the displacement of inlet.

The information about the position of actuating element overhangs to valve in the diagram in question the differential lever 7 feedback, which establish/installs the plunger of valve in the process of tracking in free position. As a result of this negative feedback, actuating element (output/yield) reproduces on the assigned scale the motion of control (inlet).

~~End Section.~~

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Page 162.

Figure 115e shows the standard block diagram of the similar automatic follower, which consists of amplifier (distributor) and hydraulic engine, which are enveloped by negative feedback. The input effect (displacement/movement), which enters the sensor (detector) of disagreement/mismatch, here is compared by equipment/device of feedback with exit effect, and on the manufactured as a result of this of comparison disagreement/mismatch between signals is establish/installed the speed of the motion of output/yield. From

Fig. 115e it follows that the input effect (displacement/movement) x, that enters on amplifier from the sensor of disagreement/mismatch, is equal to a difference in the basic input effect of the x_y, which enters this sensor, and the effect of the $x_{o.c}$, transferred from output/yield (hydraulic engine) through the feedback:

$$x = x_y - x_{o.c}.$$

Here the follow-up direct feedback in question is described by the equation, which transmits the dependence between exit and input communication/connections,

$$x_{o.c} = k_{o.c}\sigma,$$

where the $x_{o.c}$ and σ it is described exit and input feedback; $k_{o.c}$ are the gain (gear ratio) of feedback.

Are applied also the *hydraulic* boosters, in which the distribution valve (inlet) 1 is arranged directly in exit component/link (length of the lever of feedback is equal to zero). one of similar diagrams is given in Fig. 116a. Since as the bushing

of distribution valve here serves stock/rod himself 2 (exit component/link) the power piston a, system "output/yield, inlet," is covered by follow-up direct feedback.

The disagreement/mismatches, which appear between the control pressure (displacement/movement of crank 3) and the reciprocal action (output signal), are removed directly during the motion of output/yield (with its attack to inlet), i.e., it is provided the direct/straight "tracking" of output/yield (stock/rod) after the displacement/movement of inlet (plunger of valve).

Are applied also the hydraulic boosters, in which the valve is placed in the housing of cylinder (see Fig. 115c); piston rod in this case is fastened motionlessly, and mobile/motile cylinder they link with the given node/unit. Liquid in this diagram is supplied to actuating cylinder on hoses or along the axial channels, executed in motionless stock/rod.

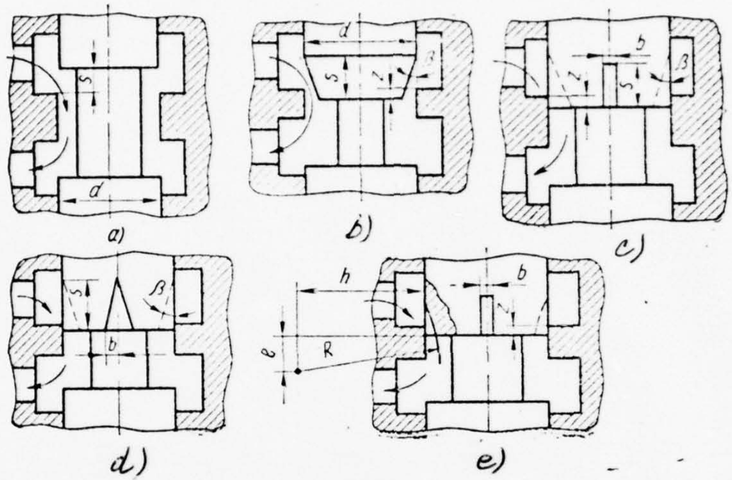
Figure 115b gives the diagram of a similar hydraulic booster without differential lever with the stepped piston of hydraulic cylinder. Piston 3 forms with cylinder 1 annular chamber b,

constantly connected with main pressure line (see also Fig. 26d), chamber a, constantly connected with gutter, and also cavity c, which with the aid of arranged/located in piston valve 2 can be connected with chambers a or b. In the position of valve, shown in Fig. 116b, cavity c is separated both from the chamber b and from chamber a. With this the closed in cavity c liquid impedes the displacement/movement of piston 2 to the left under the effect of pressure in chamber b, acting on annulus piston,

$$F=\frac{\pi}{4}(d_1^2-d_2^2).$$

In moving valve 2 to the right cavity c will be connected through the annular groove of valve with chamber b, as a result piston rod, connected with load, it will be moved under effect of pressure on the unbalanced part of it area to the right, following the valve. With the cessation of valve 2, piston, attacking to it, overlaps the slide-valve slot through which thus far they were connected chamber b and cavity c. As a result piston 3 will stop.

Fig. 117. Airfoil/profiles of the working bands of slide-valve distributors.



In moving valve 2 of this position to the left cavity c of cylinder will be connected through channel d and the slot, formed in this case by the right end/face of valve 2, with gutter, as a result piston under the effect of pressure of liquid from the side of the chamber b on annulus

$$F=\frac{\pi}{4}(d_1^2-d_2^2)$$

will move also to the left. With the cessation of valve 2, piston, attacking to valve, will overlap channel d, which connects cavity c of cylinder with overflow chamber a. As a result the piston will stop.

As it follows from diagram, the effective area of piston during its motion in both directions will be area

$$F=\frac{\pi}{4}(d_1^2-d_2^2),$$

in accordance with which of force on stock/rod

$$P = pF = \frac{\pi}{4} \Delta p (d_1^2 - d_2^2),$$

where the $\Delta p = p_{н} - p_{сл}$ — the pressure differential; here p_H and $p_{сл}$ - the pressure of forcing and gutter.

Airfoil/profile of the working bands of plunger and discharge characteristics of valve.

The sectional area of the passage channels of valve for this displacement of plunger from the mid-position and the intensity of

the increase of fluid flow rate in the course of plunger depend on the design of its working bands. In accordance with this, the value and the characteristic of discovery/opening slide valve port in the course of plunger determine in many instances accuracy/precision and the sensitivity of the hydraulic servo system.

For obtaining the maximum flow areas, the plungers usually are fulfilled with cylindrical bands and the sharp edges of both bands and the windows (circular annular grooves) in slide-valve bushing (Fig. 117a). If it is necessary to ensure a smoother change in the section of windows, through the bands of plunger, they perform with small (6-10°) conicity with certain part of the length of band (Fig. 117b) or with several slit-shaped gashes (Fig. 117c and d). From the geometric relationships of the forms of the working (throttling) cell/elements, shown in Fig. 117, it is represented possible to calculate the current working sections of passage openings depending on course s of plunger during motion to discovery/opening (beginning with zero discovery/opening) and the design parameters of valve.

Page 164.

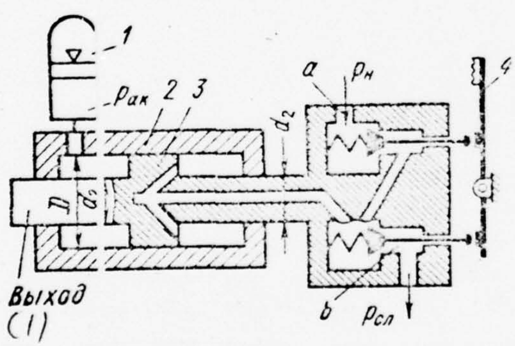


Fig. 118. Diagram of hydraulic booster with valve distributor.

Key: (1). Output/yield.

The maximum course s of valve usually is selected in the general case from the condition that the area f of discovery/opening working window would be equal to the sectional area of feeder.

Are given below the calculated dependences for valves with profiled slots (cone, flattenings and groove/slots) when the maximum piston stroke of the valve of x_{max} does not exceed the length (in the course of valve) of the shaped section of $(x_{max} \leq s)$. Sectional area f of the slots of slide valve ports can be calculated (indices of parameters f correspond to positions in Fig. 117):

$$f_a = \pi ds;$$
$$f_б = \pi (s - z)[d - (s - z)\sin\beta\cos\beta]\sin\beta;$$
$$f_в = nb(s - z)\sin\beta;$$
$$f_г = \frac{nb}{s}(s - z)^2\sin\beta;$$
$$f_д = nb\left[R - \sqrt{h^2 + (e + z)^2}\right],$$

where n they correspond the number of grooves (moustaches).

From the given equations it follows that it occurs as linear (it corresponds to the forms of slots in Fig. 117a and c), so also the nonlinear (corresponds to the forms of slots in Fig. 117b, d and e) dependence f = f (x).

Distributors of servomechanisms.

Besides slide-valve distributors (see Figs. 40a and 41) in hydraulic boosters are applied also the crane and valve distributors, structurally changed in accordance with the specific requirements for the slave/servo hydraulic drive. In automatic control systems, are applied also the distributors of the jet/stream type and type" nozzle" - shutter/valve with electric control.

Crane distributors are applied predominantly in diagrams with rotary type actuating cylinder (see Fig. 139a).

In designing crane distributor, one should provide for a decrease in friction the discharging of cock plug from the linear pressure of liquid. During the application/use of tap/cranes in servo systems, the discharging of plug is realized by the paired execution of the arranged/located diametrically opposite opening/apertures (see Fig. 54b). ; however, even, in this case crane distributors possess large friction, in view of which tap/cranes with the sliding working

cell/elements are applied usually in pressures not above 100 kgf/cm^2. For a reduction in friction the cock plug is placed in bushing on needle bearing (see Fig. 54c).

Page 165.

More rarely are applied valve distributors (see Fig. 58a), which provide because of the practical absence of dead space the high sensitivity of hydraulic booster.

Figure 118 depicts the schematic diagram of hydraulic booster with differential piston and valve distributor. The left cavity of the cylinder of 2 hydraulic boosters is constantly connected with gas-hydraulic storage battery/accumulator 1, and right with the aid of the valves, given with knob/stick 4, alternately it is connected with channel a of power supply or gutter b. In the first case piston 3 is moved to the left, displacing liquid from the left cavity of cylinder 2 into storage battery/accumulator 1, and in the second it under the effect of pressure of the liquid, displaced from storage battery/accumulator, it is moved to the right.

Since the dead space, caused by the overlap of valve distributor, here can be easily removed, this hydraulic booster differs in terms of high fidelity (tracking of output/yield after knob/stick 4). The pressure of the p_H of the power supply of system and the pressure of p_{aK} in storage battery/accumulator select on the basis of relationships (by pressure of the $p_{c\pi}$ of gutter and friction we disregard)

$$P_1 = \frac{\pi}{4}(D^2 - d_1^2)\, p_{aK\,min};$$

$$P_2 + \frac{\pi}{4}(D^2 - d_1^2)\, p_{aK\,min} = \frac{\pi}{4}(D^2 - d_2^2) p_H,$$

where P_1 and P_2 - the maximum external loads on the stock/rod of actuating cylinder during motion of it respectively to the right and to the left; D, d_1 and D_2 are diameters of cylinder and stock/rods of its piston; $p_{aK\,max}$ and $p_{aK\,min}$ are the maximum and minimum pressure and storage battery/accumulator (at the end of the piston stroke of cylinder to the left and to the right); p_H - the

pressure of the power supply (forcing) of system.

Dead zone of the slave/servo hydraulic boosters.

The important parameter, which characterizes the quality of the slave/servo hydraulic booster, is the size/dimension of dead zone (the dead space), in limits of which a change of the control signal does not produce for any reasons for the reaction (motion) of the loaded performing hydraulic engine (output/yield).

The size/dimension of this zone for an hydraulic booster with distribution valve depends first of all on the size/dimension of the overlap by the bands of the plunger of the windows of the power supply of hydraulic engine.

From the aforesaid (see Fig. 41, 43 and 115d) it follows that the band of plunger overlaps in symmetrical position with respect to these windows the appropriate window at length

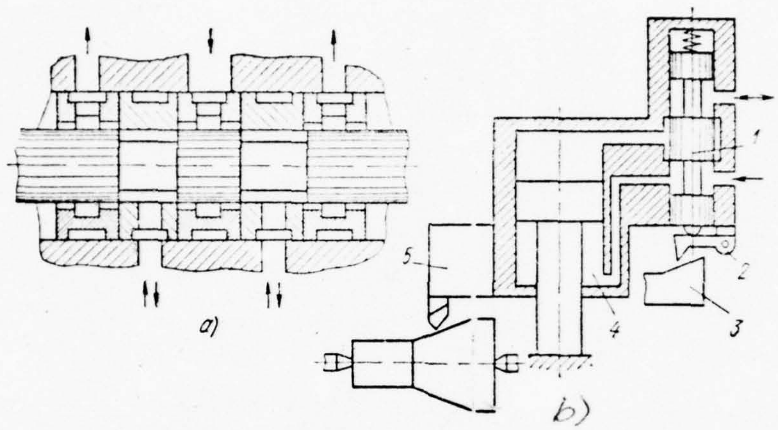
$$c = \frac{h-t}{2}.$$

In view of this during the displacement of plunger from free position to one side or the other to the size/dimension of overlap

$$c = \frac{h-t}{2}$$

the liquid into actuating cylinder (hydraulic engine) does not enter, and consequently, the displacement/movement of plunger within the limits of bilateral overlap (2c = h, t,) is not accompanied by the motion of output/yield (hydraulic engine).

Fig. 119. Valve with zero overlap (a) and circuit diagram of its into the hydraulic system of copying machine (b).



The dead zone is defined also the series other factors: by airtightness of system, by friction and by gaps of mechanical node/units, by the elasticity of system elements and working fluid, and also by the load, applied at output/yield. Gaps and elasticity in the mechanism, which links control knob with valve, increase dead zone, since the motion of setting device (control knobs) before their selection are not accompanied by feed into the hydraulic engine of energy. The load of output/yield increases dead zone, since on load depends the pressure differential of liquid in distributor. In view of this discovery/opening pressure window, necessary for the start of hydraulic engine probably taking into account the effect of the overflows of liquid into drainage cavity, the large, the large will be the this jump/drop. In transient (being unsteady) conditions can occur a supplementary change in the dead zone and the disturbance/breakdown of accuracy/precision as a result of the action of load from the pressures exerted by masses, connected to driven/known component/link, for overcoming of which will be required larger than during the steady-state mode of motion, discovery/opening the passage openings of valve.

In equal measure required discovery/opening distributor depends also on hydraulic slipes, increasing with an increase in the latter, since for their compensation is required the supplementary

expenditure. The motion of the output/yield of hydraulic engine when hydraulic slipes are present, will not be initiated until into its working chambers is given the volume of liquid, which exceeds leakages with this jump/drop in the pressure and load.

For a decrease in the dead zone, which determines the accuracy/precision of tracking, overlap $2c = h - t$ of expenditure windows in distributor are performed smallest possible, leading it to 5 μm and less up to zero ($h \approx t$). The latter include the distributors whose bushing is comprised from the pressed into housing separate rings, executed with high accuracy/precision in width (on the order of ± 1 μm). Such valves are applied in the copying machine tools, to which are presented the high requirements with respect to fidelity.

Figure 119a depicts a similar valve of the servo system of the machine tool whose bushing consists of five separate rings, two of them are utilized for the formation/education of expenditure windows even three - for central and the extreme sections of distributor. The being mated faces of sections are worked with the holding of axial size/dimension within the limits of the portion/fraction of micrometer. The hydraulic diagram of hydraulic-copying lathe with a

similar valve is shown in Fig. 119b. The support of 5 machine tool is connected with the movable cylinder of 4 hydraulic boosters. In the lower (rod) cavity of cylinder, constantly will be fed the liquid under pressure, and the upper cavity with the aid of valve 1 it is connected either with forcing or with gutter.

Page 167.

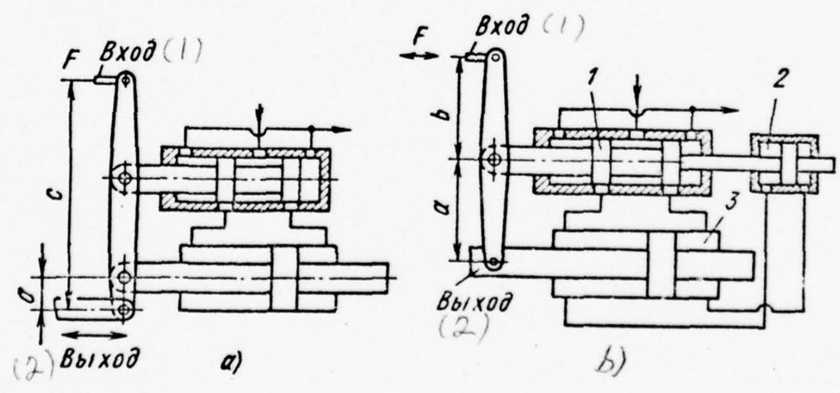


Fig. 120. Hydraulic boosters with the mechanical (a) and hydraulic (b) loading of control knob.

Key: (1) Inlet; (2) Output/yield.

Valve is moved at the signals of master form 3, which affects the plunger with the aid of probe 2, which slides on master form. During the connection/compound of the upper cavity of cylinder 4 with gutter, the support together with cutter is driven out from workpiece (billet), while during connection/compound with pressure cavity, it approaches it. The latter is caused by a difference in the effective areas of piston in the upper and lower cavities.

Mechanism of perception on the control of the force of load.

In the diagrams of servomechanism examined above operator, moving control handle, it does not perceive answer/response from the side of actuating mechanism. However, in a number of cases, it is required to ensure the perception of output/yield on control knob. This mechanism is required, for example, in the system of control linkage of aircraft or automatic machine, where during control it is usually desirable to utilize an instinctive reaction on the part of pilot or driver (in particular in emergency situations).

For this purpose the control linkage must be supplemented by the special device, which transmits part of load strength of output/yield

to control handle. This device is called the mechanism of perception.

The load of control knob can be created hydraulically and mechanical. *Figure 120a shows* the diagram of the loading of control knob of mechanical (lever/crank) method. This is reached by the fact that instead of direct (coaxial) loading to the stock/rod of hydraulic cylinder it is applied here on certain arm, thanks to which part of the load, proportional to the relation of beam arms a/c, it is transferred to control knob.

It is not difficult to see that in this case the system acquires reversibility, i.e., is provided the transmission of part of the force, developed with operator on control knob, to driven/known node/unit (output/yield), and vice versa it is provided the transmission of certain fractions of the load, which acts on output/yield, on control knob.

Force on control rod (inlet)

$$F = P\frac{a}{c} = Sp\frac{a}{c} = \frac{\pi D^2}{4}p\frac{a}{c},$$

where $S = \pi D^2/4$ and p - the piston clearance of actuating cylinder with a diameter of D the pressure of liquid.

Page 168.

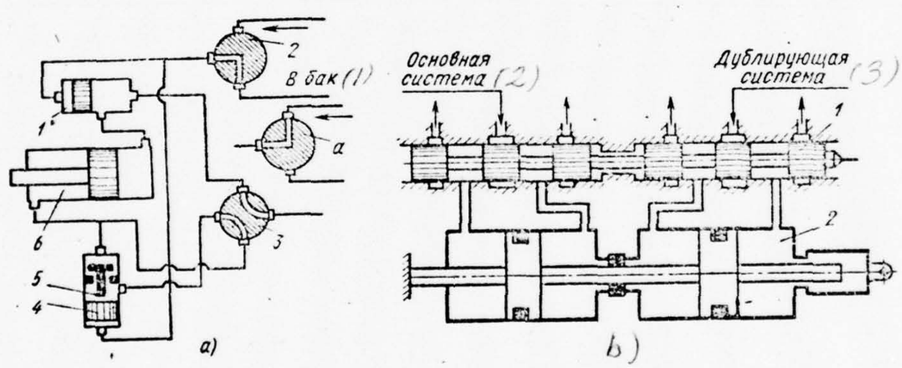


Fig. 121. Diagrams of the redundancy of control.
Key: (1) In tank; (2) Basic system; (3) Duplicating system.

Figure 120b gives the diagram of the mechanism of the perception of hydraulic action. Perception here is reached with the aid of supplementary hydraulic cylinder 2 whose stock/rod is connected with the valve of 1 control, and its cavity is connected with the corresponding cavities of actuating cylinder 3. Thus, control handle is loaded by the force, proportional the pressure differential (to load in performing by gutter cylinder 3.

Force on control rod (inlet)

$$F = Sp\frac{a}{b+a},$$

where $S = \pi d^2/4$ - the piston clearance of load cylinder 2 by diameter d.

Emergency hydraulic circuits.

In some machines as, for example, in aircraft, frequently

appears the need for having besides the basic power-supply system of performing hydraulic engine for the execution of vital functions even duplicating (emergency), which must automatically be included in the failure of the basic power-supply system. The working medium of emergency system usually is the same liquid, as basic; however, in some systems apply others, than in the basic system, liquid (for example for the emergency system of aircraft hydraulics is utilized as working medium fuel/propellant or compressed gas).

The schematic diagram of this (duplicating) system is represented in Fig. 121a. To that until is included (by hand or automatically) emergency valve 2, actuating cylinder is supplied through basic valve 3 from the basic power supply. After the setting up of emergency valve 2 at the position of power supply from emergency source (see position of a) the small piston of shuttle valve 1 (see also Fig. 90) it will move to the right, will overlap the channel of the circuit of the basic power supply and will connect the right cavity of actuating cylinder 6 with the line of emergency service.

In order to exclude the possibility of start by mistake or as a result of the malfunction of basic valve 3 to the power supply of the

opposite (left) cavity of actuating cylinder with the connected emergency service, into system is introduced the bypass valve through which the rod (left) cavity of cylinder 6 is connected with gutter. This valve is the hydraulic lock, controlled by the pressure of emergency system. During the supplying of this pressure under the small piston of 4 valves 5, it steps down and open/discloses the valve, which connects the left cavity of cylinder 6 with gutter.

Page 169.

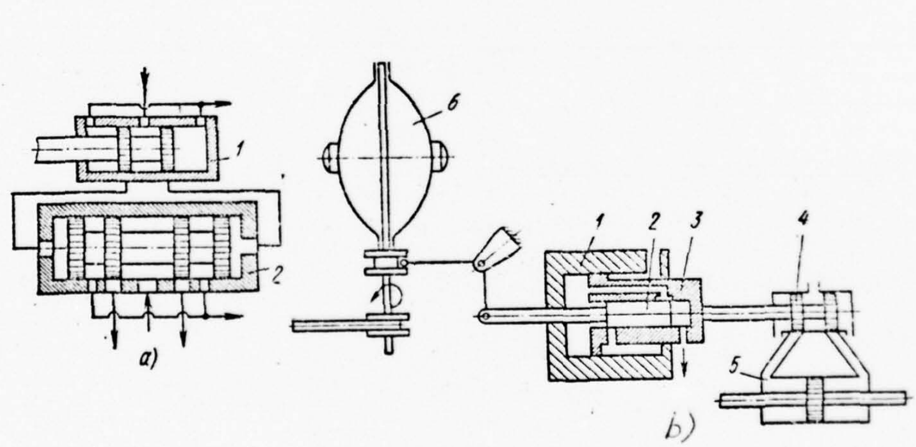


Fig. 122. Two-stage distribution valve (a) and the diagram of the two-stage hydraulic booster of control system (b).

For the same purpose is applied the dual (tandem) cylinders each of which is supplied from the autonomous hydraulic system through the dual valve with common/general/total stock/rod (Fig. 121b). Input component/link (valve) 1 hydraulic booster is connected by the system of mechanical traction, levers and cables with steering control either pilot's pedals, but exit component/link (cylinder) 2 is connected directly or through the power wiring/run with aircraft control surface. In exact system work both actuating cylinders, developing the doubled thrust. In the case of output/yield in a similar two-chamber hydraulic booster from system of one of the hydraulic systems, the second hydraulic system continues to act, providing work with that lowered/reduced 2 times of power.

When output/yield is the housing of cylinder (see Fig. 121b), valves they are placed in the latter. In this case is provided the rigid single feedback without intermediate component/links.

Two-stage slide-valve distributor of hydraulic boosters.

When control force is the signal of small power, into hydraulic drive is inserted the special hydraulic or electrical device, which

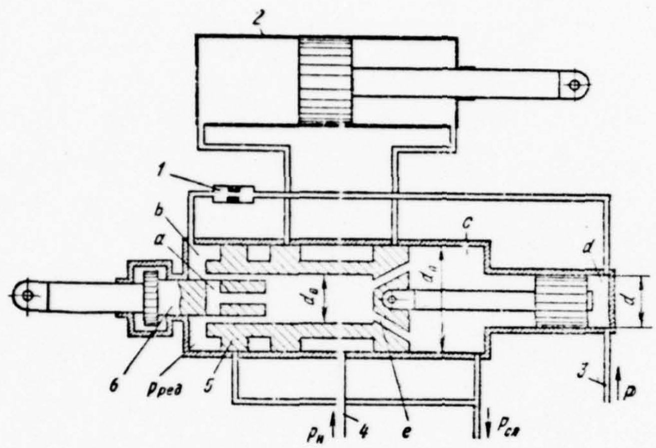
amplifies input signal to the value, sufficient for the displacement/movement of distribution valve. Hydraulic boosters with similar distribution are called hydraulic boosters with multistage amplification.

Two-stage distributor (Fig. 122a) it consists of valve 1 of first stage, to stock/rod of which is applied the control force, and valve 2 of second step/stage, with the aid of which the liquid is distributed by the cavities of performing hydraulic cylinder. Valve 1 is called valve- pilot, valve 2 - by the basic (main) valve. The exchange of basic valve 2 is realized by a working fluid at the signals of valve-pilot 1, who is the aggregate/unit, for the displacement/movement of plunger of which it is required insignificant (2-3 g) force.

Figure 122b shows the diagram of the hydraulic booster of control system with dual amplification. Ball governor 5, connected with the output shaft of the controlled machine, affects during a change in the speed of this shaft to plunger 2 of auxiliary (pilot) valve of first stage of intensification, which controls piston 3 of auxiliary actuating cylinder 1 of second step/stage of intensification, and the latter controls basic distribution valve 4,

which feeds power (performing) cylinder 5 affecting the control of machine.

Fig. 123. Diagram of hydraulic booster with dual amplification.



The diagram of analogous system with dual amplification is represented in Fig. 123. The plunger of 5 basic distribution valve is found p of the liquid, applied by the booster pump through conduit/manifold 3 into the right cavity d of cylinder, and under the reduced pressure of $p_{peд}$ in the left cavity b into which liquid enters from conduit/manifold 3 through throttle/choke 1 (cavity c of this cylinder is connected with tank). Since the area of plunger is less than the area of plunger 5, latter under the effect of pressure of liquid is wrung out to starboard, open/disclosing in this case the passage of liquid from working main line 4 into the right cavity of actuating cylinder 2.

In the axial opening/aperture of plunger 5, enters the plunger of 6 auxiliary valve, with extraction of which to the left left cavity b of the cylinder of the basic valve is connected through channels a and e in plunger 5 with drain line. Since the supply of liquid into the left cavity b is conducted through throttle/choke 1, pressure in this cavity depends on the position of plunger 6. From diagram it follows that during the displacement of the latter to the left plunger 5 under the effect of pressure of liquid on small piston with a diameter of d will move also to the left, after connecting in this case the left cavity of actuating cylinder 2 with working main line. With the cessation of plunger 6, plunger of 5 basic valve,

moving, will overlap channels a, in consequence of which the pressure of the liquid, which enters the left cavity of cylinder b of the valve through throttle/choke 1, will be raised to the value, capable of balancing pressure on small piston, whereupon the motion of plunger 5 will cease itself.

The displacement of the plunger of 6 auxiliary valve is accompanied to the right by the overlap of channels a, as a result of which pressure in the left cavity of cylinder b will be raised and will move plunger 5 to the right. Consequently, the plunger of 5 distribution valve will repeat the motions of the plunger of 6 auxiliary valve.

Throttle/choke 1 besides its basic designation/purpose (creation the pressure differentials in the cavity of cylinder b of distribution valve) serves also as the damper, which prevents the fluctuations of plunger 5.

Page 171.

The equilibrium condition of the static forces, which act on plunger 5, can be presented in the form (pressure in drainage line from cavity c we disregard)

$$pf_1 = p_{ped} f_2,$$

where p - the pressure of liquid in the right cavity of the cylinder of small piston; $f_1 = \pi d^2/4$ - the effective sectional area of small piston; p_{ped} are the reduced pressure in the left cavity of the cylinder of the basic valve; $f_2 = \frac{\pi d_n^2 - d_s^2}{4}$ — the effective sectional area of the left end/face of plunger 5; here d_B - the diameter of the plunger of 6 auxiliary valve.

It is obvious that with $f_2 > f_1$ the equality of the forces, which act on plunger 5, will begin under condition

$$p_{ped} = p\frac{f_1}{f_2}.$$

With $p_{ped}>p\frac{f_1}{f_2}$ plunger 5 will be moved to the right, while **with** $p_{ped}<p\frac{f_1}{f_2}$ — to the left.

The displacement/movement of plunger 5 both in that and in another case will continue until the reduced pressure into cavity b, which is changed during the displacement of plunger, caused by the displacement of the plunger of 6 auxiliary valve, achieves the value at which will be establish/installed the equality

$$p_{ped}=p\frac{f_1}{f_2}.$$

The minimum diameter of plunger 6 auxiliary (pilot) valve of d_B is selected depending on the technological possibilities of its production. Virtually it is usually equal to 2-3 mm. For decrease the silt of friction of the plunger of 6 auxiliary valve the power supply of the system of its control usually is realized from auxiliary power device with low pressure (3-5 kgf/cm²).

Follower with the tracking bushing of servo-valve.

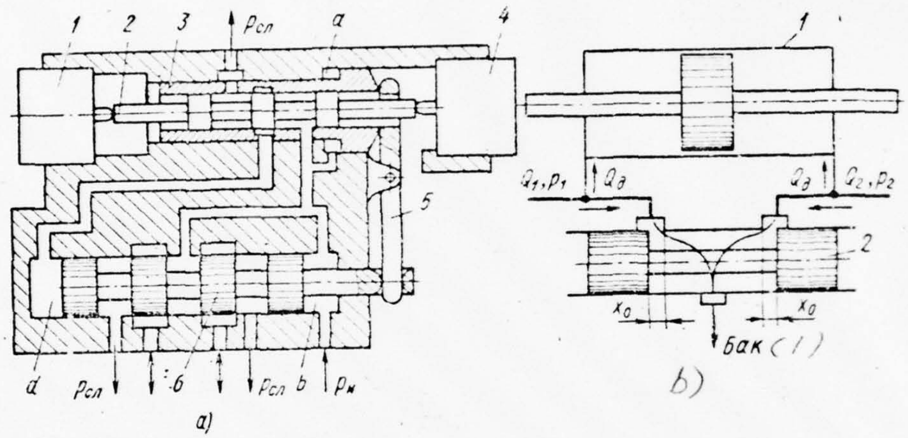
In many instances are applied the diagrams in which output/yield connect through feedback not with the plunger of distributor, but with its bushing, by displacement of which through feedback relative to plunger is removed the introduced during control disagreement/mismatch.

Figure 124a depicts the diagram of this follower of the two-stage type, valve 2 of first stage of intensification of which (servo-valve) is controlled with the aid of two electromechanical converters 1 and 4. Feedback in this mechanism is realized through rocker arm 5, upper end of which is connected with floating hub 3 servo-valve (by inlet) and butt end - with the plunger of 6 basic valve (by output/yield). Bushing 3 servo-valve by the pressure of the P_H of the forced liquid, applied into the chamber, constantly is adjusted to upper end of lever 5, duplicating the motions of the latter.

Let us assume that the plunger of valve 2 with the aid of left converter is displaced from free position to starboard to the size/dimension, which exceeds overlap. In this case the end-type chamber d of basic plunger 6 will be connected through the discovered slot of the window of servo-valve with drain channel, as a result the plunger of 6 basic valve under the effect of pressure of the P_H of liquid on chamber b will be displaced toward left side, after connecting in this case with pressure channel the appropriate cavity of hydraulic engine. Since floating hub 3 is connected by the force of the pressure of liquid with lever 5, connected with plunger b, the displacement of the latter to the left will cause the displacement of bushing 3 to the right, i.e., bushing will follow plunger 2.

Fig. 124. Mechanism with the tracking bushing (a) and power supply from two sources (b).

Key: (1). Tank.



In such a case, when the displacement/movement of plunger 2 ceases itself, the incoming bushing will overlap the window of the power supply of chamber d, after fixing plunger 6.

During the displacement of plunger, 2 servo-valve to left side process flow/lasts in the backward sequence: chamber d will be connected with the forcing main line, and the plunger of 6 basic distributor will move as a result of a difference in the effective areas of its left and right sides to the right, changing over the power supply of hydraulic engine and simultaneously displacing through lever 5 bushing 3 to the left.

Hydraulic boosters with two power supplies.

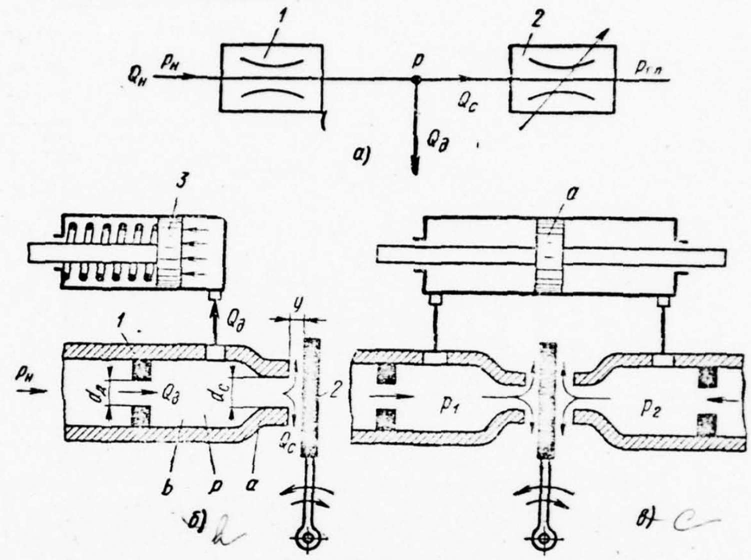
Is of practical interest the diagram of hydraulic booster with flowing (negative overlap) valve and power supply from two independent sources (pumps) with the constant feed $Q_1 = Q_2 = const$ (Fig. 124b). The controlled throttle/chokes in the form of the edges of plunger 2 are included in this diagram in parallel to hydraulic engine (cylinder) 1. In the mid-position of plunger 2 with the symmetrical location of the windows of pressure in the right (p_2) and

left (p_1) cavities of cylinder 1 are equal to ($p_1 = p_2$) and its piston is found in rest. In this case, the liquid in volume $Q_1 + Q_2$, which enters from power supplies through windows with the initial discovery/opening x_0, is directed to for tank. During the displacement of plunger (during the supplying of signal) of mid-position one of the passage openings will increase, and the second decreases, as a result the equilibrium of system will be destroyed ($p_1 \neq p_2$). In this case, part of the fluid flow rate from one of the sources (from the side of the reduced expenditure window) will enter the cylinder whose piston will arrive to motion. The direction and the speed of motion are determined by sign and the displacement of the plunger of 2 valves. During the displacement of plunger 2, for example, to left side to size/dimension, the equal or exceeding initial clearance x_0, feed Q_2 the right power supply will enter the right cavity of actuating cylinder 1, and feed Q_1 the second source will enter the tank. This position of plunger corresponds to the maximum speed of the piston stroke of actuating cylinder to left side $v = Q_2/F$ and maximum to the developed with it force $P = p_2F$, where F correspond the effective area of cylinder. The flow rate of the second power supply is directed to for tank with the pressure, equal drainage.

The hydraulic boosters of this diagram differ in terms of high

sensitivity; however, require the application/use of two independent power supplies, and also they have the large overflows of liquids, which in the mid-position of valve (during the zero bias of its plunger) are equal to the total feed of both pumps. In view of this the hydraulic booster of this diagram is applied only in small powers.

Fig. 125. Amplifiers of the type" nozzle" - shutter/valve.



Hydraulic boosters of the type "nozzle "- shutter/valve.

In servo systems, especially in automatic control systems, are used extensively choke hydraulic boosters of the type" nozzle" - shutter/valve. They are applied in essence in electrohydraulic two-stage systems as first stage of the intensification of input signal and thinner - for an immediate effect for the controlled objective.

The mechanism of amplifier nozzle - shutter/valve is the hydraulic potentiometer, which consists of two hydraulic resistance (Fig. 125a), from which resistance 1 is constant and resistance 2 adjusted in value. Cavity (chamber) between throttle/chokes with pressure p is connected with hydraulic engine.

Figure 12b depicts the circuit of this amplifier, used for a control of the actuating cylinder of one-sided action. Amplifier consists of nozzle a and the powerful to be moved relative to it shutter/valve (plates) 2, the alternating/variable slot y between which is the adjustable throttle/choke.

The throttle/choke of 1 fixed resistor is carried out in the form of washer with opening/aperture. By means of the rotation of movable shutter/valve 2 it is possible to change distance y between the nozzle and the shutter/valve (to overlap the outlet of nozzle a), regulating thereby fluid flow rate from the interthrottle chamber b, connected decrease in clearance y of pressure in chamber b higher than the value, determined by compression of spring and by the external load of the stock/rod of hydraulic engine 3, the latter will be moved, compressing spring. During a decompression, the spring will return the piston of hydraulic engine 3 at starting position. Shutter/valve is fastened on the being turned lever with sufficiently large arm, that makes it possible to consider its displacement/movements relative to nozzle forward/progressive, and the slot between them - parallel.

Page 174.

The expenditure/consumption of $Q_н$, applied to the throttle/choke of 1 fixed resistor, is divided into two flows of $Q_д$ and $Q_с$, relative values of which depend on the resistance of the

adjustable throttle/choke, determined by the position of shutter/valve 2 relative to nozzle edges a (from the area of slot nozzle - shutter/valve), the flow of Q_∂ entering performing hydraulic engine (cylinder) 3 and the flow of the Q_c through the adjustable throttle/choke to gutter.

The dependence of pressure p in the interthrottle chamber b (and respectively in hydraulic engine 3) from the resistance of the adjustable throttle/choke, serves as the power characteristic of the hydraulic potentiometer in question.

In throttle/choke nozzle - shutter/valve the flow of liquid and hydraulic resistance in essence are caused by the same physical phenomena, as are caused by sudden flow expansion after its contraction in end-type slot and by sharp incidence/drop the speeds of liquid in the zone of detached flow. This flow expansion is accompanied by intense vortex formation and braking by liquid in the chamber at output/yield from the slot which can be assume/taken practically motionless.

With the condition of the $Q_\partial = 0$ of loss of pressure on

throttle/chokes it is possible to present (pressure of the $p_{сл}$ of gutter we disregard):

$$\Delta p = p_{н} - p = \frac{Q_c^2}{G_{\partial p}^2}$$ - for uncontrolled throttle/choke 1;

$$p = \frac{Q_c^2}{G_c^2}$$ - for adjustable shutter/valve 2,

where the $G_{\partial p} = \mu_{\partial p} A_{\partial p} \sqrt{\frac{2}{\rho}}$ is hydraulic conductivity of uncontrolled throttle/choke 1; $G_c = \mu_c A_c \sqrt{\frac{2}{\rho}}$ - the hydraulic conductivity of valve 2; $A_{\partial p} = \frac{\pi d_{\partial p}^2}{4}$ and $A_c = \pi d_c y$ - the sectional area of throttle/choke 1 and of valve 2;

here $\mu_{\partial p}$ and μ_c - the coefficients of the expenditure/consumption of throttle/choke 1 and of shutter/valve 2; y - distance (width of slot) between the nozzle edge and the shutter/valve.

For the displacement/movement of shutter/valve, are required negligibly the small effort/forces, which can be created by the setting device of insignificant power - by ball governor, by low-power electromagnet, the pressure sensors and temperatures and etc. These amplifiers allow/assume with the slow speeds of shutter/valve to 30 inclusions per second.

The mechanism of the action of an amplifier of this type let us examine in an example of its application/use in the automatic control system of the speed of steam turbine, realize/accomplished with the aid of rotary steam valve 1 (Fig. 126a). during the supplying of liquid under certain constant pressure p_0 = const on the entrance of the first resistance (uncontrolled throttle/choke) 6 we can obtain by changing the position of shutter/valve 3 relative to nozzles 2 variable pressure p in the chamber before nozzle 2, connected with performing hydraulic cylinder 7. With an increase in the frequency of the rotation of the adjustable object, the governor balls 4 diverge, moving by means of clutch 5 shutter/valve 3 to the side of the overlap of nozzle hole. In this case, the pressure in the interthrottle chamber is raised, and hydraulic cylinder 7 control of steam valve ease pair.

For an increase in the stability in transient conditions the examined circuit they supply with the supplementary auxiliary hydraulic cylinder of 1 pressure feedback (Fig. 126b). The spring-opposed piston rod of this cylinder is connected with the second (upper) end of shutter/valve 4 which, as in the preceding/previous circuit, is controlled by the ball governor,

connected with it with thrust/rod 3.

Page 175.

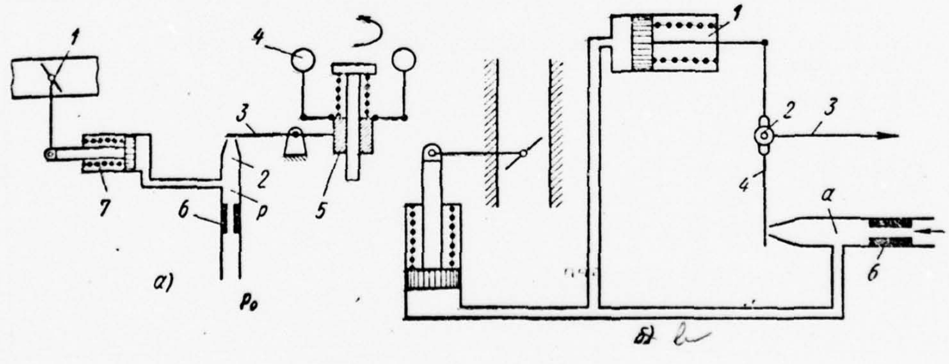
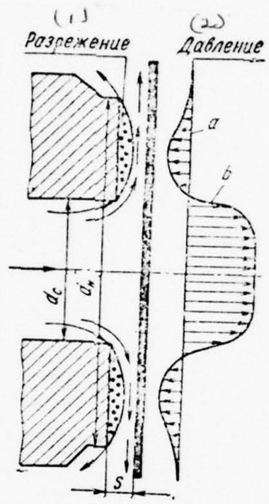


Fig. 126. Standard diagrams of systems of the type" nozzle" - shutter/valve.

Fig. 127. Action of fluid flow to shutter/valve.

Key: (1). Evacuation/rarefaction. (2). Pressure.



Since upper end of shutter/valve 4 is connected with the stock/rod of hydraulic cylinder 1, it additionally is controlled also by the piston of this cylinder.

From diagram it follows that any change in the position of shutter/valve 4 will cause pressure change in the interthrottle chamber a. Since this chamber is connected with the left cavity of hydraulic cylinder 1, this pressure increase will cause the displacement of piston and, consequently, also the displacement of upper end of the shutter/valve 4 relative to axle/axes 2 its rotations, as a result the butt end of the shutter/valve will be moved at starting position with respect to nozzle. The characteristic of amplifier, influence some design parameters, in particular the width of nozzle edge. With a known increase in this width (with an increase in the relation of the $d_н/d_c$, where the $d_н$ and d_c are the outer diameter of the end/face of nozzle and the inner diameter of opening/aperture cap/filling) is observed the disturbance/breakdown of the stability of the power reaction of flow to shutter/valve, especially with small end-type clearances. Under some conditions, which depend on the mode/conditions of discharge, in clearance (slot) between the nozzle edge and the shutter/valve is formed the rarefaction zone. From Fig. 127 in which is represented the diagram of the pressure of flow to shutter/valve (curve b) it is

apparent that the latter in rarefaction zone will be located lower than the pressure in the flooded medium (is curve a), into which escapes the liquid from nozzle, that it can lead to a sign change of the effort/force of the reaction of fluid flow to shutter/valve which in this case will act in the direction of a decrease in the clearance between the nozzle edge and the shutter/valve. With wide nozzle edge, is disrupted also as a result of the effect of viscosity change the stability of the escape of liquid.

For a decrease in the effect of the pressure of liquid on the surface of shutter/valve, and the also for the stabilization of this effect during the displacement/movements of shutter/valve (during change y) one should decrease the width of the $s=\frac{d_{н}-d_{c}}{2}$ of nozzle edge. This requirement is dictated by the tendency to decrease the viscosity effect of liquid on system performance. With narrow edges to nozzle it is possible to use the laws of flow of liquid through the opening/aperture in fine/thin wall. Of practically the width of the $s=\frac{d_{н}-d_{c}}{2}$ of the edge of nozzle edge must be less than the minimum clearance y between the nozzle edge and the shutter/valve of $s<y_{min}$.

Page 176.

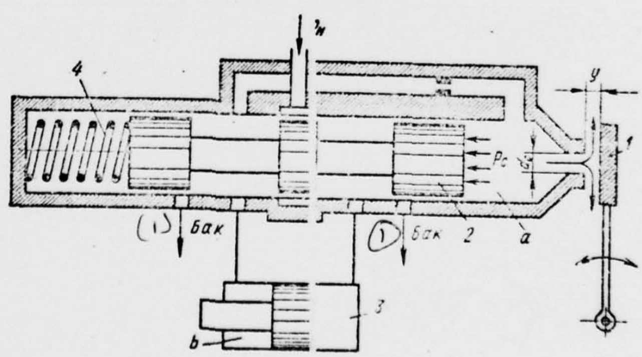


Fig. 128. A two-stage amplifier of the type" nozzle" - shutter/valve.
Key: (1) Tank

On the other hand, the maximum clearance of y_{max} must be less than the $d_c/5$, where the d_c - the diameter of opening/aperture and aaaa is a outside diameter of nozzle. Practically the diameter of outer circumference of the edge of nozzle is equal $d_н = (1,2 \div 1,5)\, d_c$, the diameter of shutter/valve is equal $(3-4)\, d_c$.

For a control of the actuating cylinder of bilateral effect, is applied the diagram with two nozzles (see Fig. 125b) in which is reached the considerable discharging of shutter/valve from the statically unbalanced dynamic pressure forces of liquid jets. In this case the unbalanced effort/forces on shutter/valve will appear only with the jump/drop in the nozzle pressures, caused by the displacement/movement of shutter/valve.

This device (hydraulic bridge) it consists of two hydraulic potentiometers with four hydraulic resistance two of which constants and two - variables. The hydraulic conductivity of both adjustable throttle/chokes nozzle - shutter/valve changes simultaneously during the displacement of shutter/valve from its free position: during the displacement of shutter/valve, the to the left hydraulic conductivity of the left slot of throttle/choke decreases and pressure p_1 in the interthrottle chamber increases; simultaneously the conductivity of

the right slot of throttle/choke increases, and pressure p_2 decreases. Thus, is created the pressure differential $\Delta p = p_1 - p_2$, the greater, the greater the displacement of shutter/valve from its free position.

In accordance with this, is represented possible to ensure the displacement/movement of the piston of the actuating cylinder, controlled by this device, the practically directly proportional to the displacement of shutter/valve.

Two-stage amplifiers.

For an increase in the sensitivity of amplifier and provision simultaneously for an increase in the power of output signal, are applied the two-stage devices, by first stage of intensification of which is usually the amplifier of the type" nozzle" - shutter/valve and the second, i.e., valve. The schematic diagram of this device is shown in Fig. 128. The interthrottle chamber a of this diagram is connected with the right cavity of the basic distribution valve plunger 2 which is located in equilibrium under the action of the effort/force of spring 4 and of the pressure of liquid on this

chamber. Liquid constantly will be fed into the rod cavity b of the actuating cylinder whose piston with the simultaneous supply of liquid into opposite cavity is moved as a result of a difference in the piston clearances to the left, and during the connection/compound of this cavity with tank - to starboard.

Fig. 129. Servo systems with two-stage distributors of the type" nozzle" - shutter/valve and pressure feedback.

Key: (1). Entrance. (2). Output/yield.

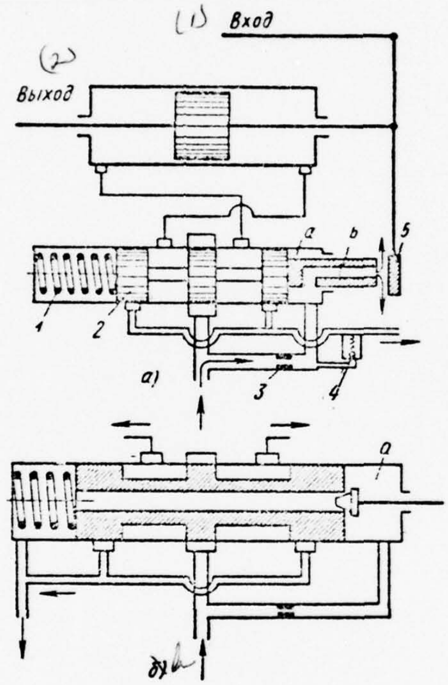


Figure 128 amplifier shows in the free position in which the right cavity of cylinder 3 is overlapped. During the displacement of shutter/valve, 1 equilibrium of forces, which act on the plunger of 2 valves, will be destroyed, and it, being displaced to the appropriate side, it will connect the right cavity of actuating cylinder 3 either with the cavity of supply (pressure of p_K), or with tank. Because of the fact that the effort/force, created a pressure of liquid on the plunger of 2 valves, is balanced by spring 4, the displacement/movement of distribution valve will be proportional displacement/movement the proportionality of fluid flow rate through the valve and the displacement/movements of shutter/valve. Consequently, in this case occurs pressure feedback.

The schematic of the application/use of this distributor in servo system is given in Fig. 129a. The plunger of valve 2 in this schematic is located in equilibrium under the action of the effort/force of spring 1 and of the pressure of liquid on chamber a, which is connected with feed line through throttle/choke 3 and with gutter - through boring b in piston rod. The resistance of the last/latter channel and, consequently, also pressure in chamber a can be changed with the displacement of shutter/valve 5; in this case as a result of the disequilibrium of the tensile stresses of spring and pressure of liquid, the plunger of slide valve will follow the

shutter/valve. For an increase in the sensitivity, the pressure in chamber a usually is reduced with the aid of valve 4 to 2-3 kgf/cm² or by means of the supply of this chamber from separate source and, in particular, from drain line.

The variety of the last/latter schematic is the schematic, presented in Fig. 129b (see also Fig. 123), in which the throttle plate, carried out in the form of conical needle, is established/installed on the entrance into chamber a. The diameter of the locking part of the needle usually does not exceed the fractions of millimeter. In the last/latter schematic is represented possible to practically remove the action on the shutter/valve (needle) of the unbalanced forces of the pressure of liquid, thanks to which for its displacement/movement in the absence of gaskets they are required negligibly small effort/forces - the order of several grams.

The minimum diameter of the nozzle hole of the first cascade/stage of intensification frequently is led to 0.1-0.15 mm. In this case, the clearance between the nozzle and the shutter/valve by operating position usually does not exceed 0.025 mm. The diameter of the aaaaa of the opening/aperture of the uncontrolled throttle/choke usually 2 times is more the diameter of the aaaa of nozzle hole.

Power effect of jet on shutter/valve.

During calculations of the amplifiers of systems in question frequently is of interest the effect on the shutter/valve of liquid jet whose value in a number of cases can turn out to be that commensurable with the effort/force, developed with the cell/element, governing the displacement/movement of shutter/valve.

Specifically, this effect can destroy the balance of the forces, which act on the armature of governing electromechanical converter and consequently, it can destroy control of the servo hydraulic drive.

Page 178.

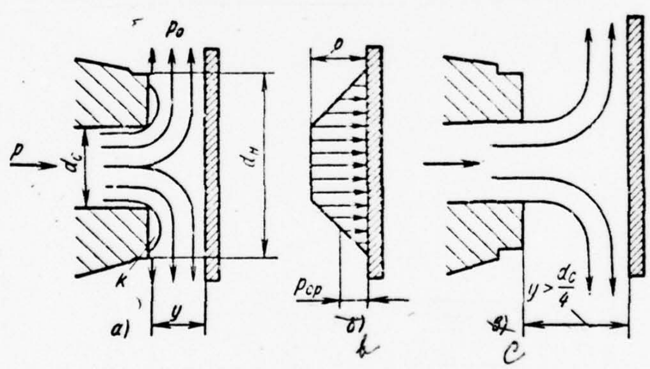


Fig. 130. Schematics of the action of fluid flow to shutter/valve.

The hydraulic reaction of the jet, which escape/ensues of nozzle, can also cause appearance in the electrohydraulic amplifier of negative feedback.

The power effect of jet on slide depends on the mode/conditions of the escape of liquid from nozzle and the design features of pair nozzle - shutter/valve.

During small discovery/openings of the nozzle of the $\left(y<\frac{d_c}{4}\right)$, which correspond to operating conditions, occurs nonseparable flow conditions, which is characterized by the fact that the flow fills entire clearance (Fig. 130) and liquid flows between the nozzle-exit plane and the shutter/valve as in flat/plane capillary clearance. During the rotation of flow into this clearance and the flow about the right angle with unrounded edges, appears the local separation of flow and the range of turbulence k; however, further flow is expanded and acquires the thickness, equal to distance y between the nozzle edge and the shutter/valve.

With an increase in discovery/opening the $\left(y>\frac{d_c}{4}\right)$ of nozzle (Fig. 130c) nonseparable mode/conditions transfer/converts to

separating, that is characterized by the fact that in jet after the passage of the edge of nozzle is establish/installed the pressure, equal to ambient pressure. Liquid in this case is not contacted with the plane of the edge of nozzle, but being hit against shutter/valve, it spreads on it in radial directions, in view of which a further increase in gap y by expenditure/consumption in practice does not influence - the expenditure/consumption in this case is determined by the section of the outlet of nozzle.

Experience shows that the shutter/valve effectively throttles under condition

$$y < \frac{d_c}{5}.$$

in view of which the maximum course of shutter/valve usually they limit by value

$$y = \frac{d_c}{5}.$$

During the nonseparable flow of liquid of $(y < 0{,}25 d_c)$ the effort/force on shutter/valve is created by the flow forces, caused by change in the momentum of

$$P_z = \rho Q v,$$

and by force P of the static pressure of liquid on the area of shutter/valve, equal to the area of the external section of nozzle edge.

Page 179.

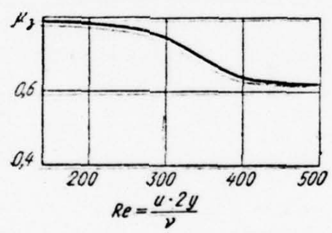


Fig. 131. Dependence of the coefficient of the expenditure/consumption of a throttle/choke of the type" nozzle" - shutter/valve on Re.

By disregarding a pressure drop in range k, and also by the turbulence of flow and by pressure in overflow chamber, we will obtain the effort/force of the static pressure of liquid on shutter/valve

$$P=\frac{\pi d_c^2}{4}p_c+\frac{\pi(d_н^2-d_c^2)}{4}p_{cp}=\frac{\pi}{4}\left[d_c^2p_c+(d_н^2-d_c^2)p_{cp}\right],$$

where the $d_н$ and d_c - external and nozzle bore; p_c - pressure before the nozzle (in the interthrottle chamber); p_{cp} are the medium clearance pressure between the nozzle edge and the shutter/valve.

Assuming that pressure in the end-type clearance between the nozzle and the shutter/valve on a radius from $\frac{d_c}{2}$ to $\frac{d_н}{2}$ (Fig. 130b) it changes according to linear law, we have

$$p_{cp}=\frac{p_c}{2}.$$

Under this condition the effort/force of static pressure on shutter/valve

$$P = \frac{\pi}{2} p (d_{н}^{2} + d_{c}^{2}).$$

The In addition to these forces on shutter/valve act flow forces; however, the calculations show that at the widespread value of the $y < 0{,}25 d_{c}$ of hydrodynamic force is small in comparison with the force of hydrostatic pressure, in view of which practical calculations in the majority of cases are conducted taking into account only static pressure.

Fluid flow rate.

Fluid flow rate through the throttle/choke of adjustable resistor will be determined in essence by the resistance of the end-type slot between the nozzle edge and the shutter/valve. The flow rate of the Q_{∂} of the liquid through the opening/aperture of the throttle/choke of the fixed resistor by the diameter of $d_{\partial p}$ is determined according to expression (26).

Assuming that the fluid flow rate to the controlled performing cylinder is equal to zero, and assuming that during the small displacement y of shutter/valve area f of the cross section of the end gap between the nozzle section and the shutter/valve is equal to the area of the lateral surface of a cylinder with height y and a diameter equal to the internal diameter d_c of the section of the nozzle, i.e., assuming $f = \pi d_c y$ it is possible to approximate the expression for flow through this gap;

$$Q_{вac} = \mu_з \pi d_c y \sqrt{2 \frac{p_c - p_0}{\rho}}, \qquad (62)$$

where the $\mu_з$ - the coefficient of the flow rate of the adjustable throttle/choke nozzle - shutter/valve (slot, formed by the nozzle edge and by shutter/valve), which is function Re; the form of the function for the amplifiers of the widespread size/dimensions is shown in Fig. 131; p_0 is pressure after shutter/valve (in the medium into which occurs the escape of liquid).

From the given in Fig. 131 curve it follows that in the range of linear law (Re > 400) it is possible to set/assume

$$\mu_з = \text{const} \approx 0{,}62.$$

Page 180.

Converting equation (62) taking into account the area of nozzle hole

$$f_c = \frac{\pi d_c^2}{4},$$

we will obtain

$$Q_{зас} = 4\mu_з \frac{d_c}{d_{\partial p}} \cdot \frac{y}{d_{\partial p}} f_c \sqrt{2\frac{\Delta p}{\rho}} = \mu' f_c \sqrt{2\frac{\Delta p}{\rho}},$$

where the $\mu' = 4\mu_з \frac{d_c}{d_{\partial p}} \cdot \frac{y}{d_{\partial p}}$ - the coefficient of the flow rate of nozzle - shutter/valve, referred to the constant area of nozzle hole.

Since the coefficient of the flow rate of the $\mu_з$ through the clearance of nozzle - shutter/valve during nonseparable flow (see Fig. 129a) practically is constant and does not depend on Re (see Fig. 131), coefficient μ' it depends in this mode/conlitions practically only on the relative clearance of y/d_c.

Experience/experiment shows that the maximum coefficient of the flow rate of aaaa occurs with clearance

$$y=\frac{d_c}{4},$$

i.e. with this clearance when the flow passage cross-sectional area of the end-type slot of $f=\pi d_c y$ is equal to the sectional area of the outlet of nozzle

$$f_c=\frac{\pi d_c^2}{4}.$$

Under this condition the range of the displacement of shutter/valve relative to nozzle edge

$$0<y<\frac{d_c}{16}.$$

Under the previously accepted condition of zero flow rate on the performing cylinder (hydraulic engine) I could write on the basis of the equality of the flow rates through the throttle/chokes of constant and the variable of sections

$$\mu_{\partial p}\frac{\pi d_{\partial p}^2}{4}\sqrt{\frac{2(p_H-p)}{\rho}}=\mu_{\vartheta}\pi d_c y\sqrt{\frac{2(p-p_0)}{\rho}}.$$

Hence the pressure of $p_{c\pi}$ in overflow chamber (allow/assuming

$p_0 = 0$)

$$\mu_{\partial p} d_{\partial p}^2 \sqrt{p_н - p} = 4\mu_з y d_c \sqrt{p},$$

$$p = p_н \frac{1}{1 + 16\left(\frac{d_{\partial p}}{d_c}\right)^2 \left(\frac{y}{d_c}\right)^2 \left(\frac{\mu_c}{\mu_{\partial p}}\right)^2}.$$

With y = 0 (nozzle is enclosed) $p = p_н$ and with y = ∞ pressure p = 0.

The obtained equation expresses the law of pressure change in the intermediate (interthrottle) chamber of amplifier in the function of the displacement of shutter/valve.

The resistance of the very nozzle hole by the diameter of d_c is selected so that it does not considerably affect the total

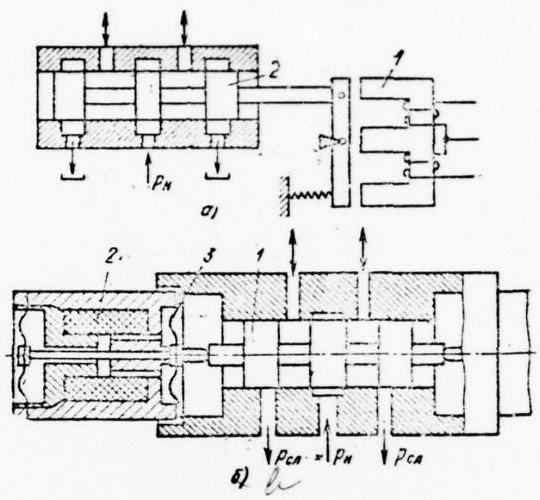
resistance of the adjustable throttle/choke nozzle - shutter/valve. Practically the resistance of this opening/aperture must not exceed approximately 10o/o of adjustable hydraulic resistance by end-type clearance (slot) between the nozzle edge and the shutter/valve. For this provision must be observed condition

$$\frac{f}{f_c} < 0{,}25,$$

where the $f = \pi d_c y$ - the area of the slot between the nozzle and the shutter/valve, calculated from the lateral surface of nozzle the diameter of d_c and by height/altitude y; $f_c = \frac{\pi d_c^2}{4}$ - the sectional area of nozzle hole.

end section.

Fig. 132. Distributors with electrohydraulic control.



SERVO ELECTROHYDRAULIC SYSTEMS.

In many instances of applying servo systems, and in particular in automatic hydraulic equipment input signal is first stage of the intensification of signal. As a result the hydraulic system is converted into the electrohydraulic servo system.

In the systems of automatic hydraulic equipment, are applied predominantly the hydraulic boosters with control with the aid of the electromechanical devices which convert output signal on position into electrical signal and, comparing the latter with signal at entrance, they affect in accordance with a difference in these signals the flow regulator of liquid.

For the transformation of input electrical signal into the displacement of hydraulic distributor, apply one- two-stage (stepped) converting devices.

In the schematic of one-stage device (Fig. 132a) electromagnet 1 directly moves the plunger of valve 2 to the size/dimension,

proportional to a change in the force cf electric current. Similar one-stage devices are applied in the small flow rates (8-10 l/min) of liquid.

Figure 132b gives the schematic of the drive of the miniature valve (diameter of plunger is equal to 3-5 mm) of the forward/progressive action of the electric converter of the rectilinear motion, which consists of two solenoid coils 2, armature travel 1 which is balanced by two leaf springs 3. Because of the application/use of two springs, which affect the plunger from two sides, is ensured the displacement of plunger, directly proportional to the current strength of control.

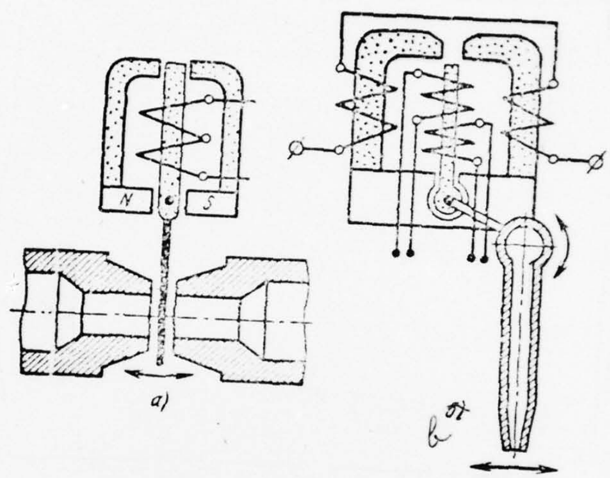
For an increase in the reliability, works to the valve of converter usually communicate oscillating motions. For this purpose into control winding simultaneously with governing electrical signal is supplied the oscillatory signal in the form of the harmonic oscillations of current with small amplitude, but with high frequency (100-200 Hz).

The servo hydraulic drive with electrical control is carried out

usually in the form of the multistage automatic control system in which input electrical signal is amplified and is converted by electromechanical converter into the displacement of the distributor, governing performing hydraulic engine. In automatic systems as first stage the intensifications usually serve nozzle - shutter/valve (see Fig. 125) or jet pipe (see Fig. 153).

Electromechanical converter is carried out in the form of the electromagnet of the rectilinear either rotary motion, mobile/motile armature of which strictly is connected with shutter/valve (Fig. 133a) or with jet pipe (Fig. 133b).

Fig. 133. Schematics of electromechanical converter.



The operating principle of this electromechanical converter is based on the interaction of two magnetic fluxes, created by the currents, which take place on the excitation windings and control. In the case of the equality of these current strength, the magnetic flux of control will be equal to zero. During a change in the current strengths, appears the magnetic flux, proportional to a difference of these current strength, under action of which armature, and together with it and shutter/valve they are moved. The developed in this case at the armature of electromagnet driving/moving effort/force is balanced by the effort/force of antagonistic spring. During the disagreement/mismatch of these effort/forces, the armature and shutter/valve are deflect/diverted to the size/dimension, directly proportional to the current strength of control.

Converters with an hydraulic booster of the type" nozzle" - shutter/valve differ in terms of high speed operation($\leq 0.5 \cdot 10_{4}$ s) and in terms of a comparatively large thrust; however, they can ensure only the small armature travel and shutter/valve (≤ 0.15 mm).

In the two-stage electrohydraulic amplifier, presented in Fig. 134, first stage of amplification is carried out in the form of nozzles 3 and of shutter/valve 2, controlled with the aid of

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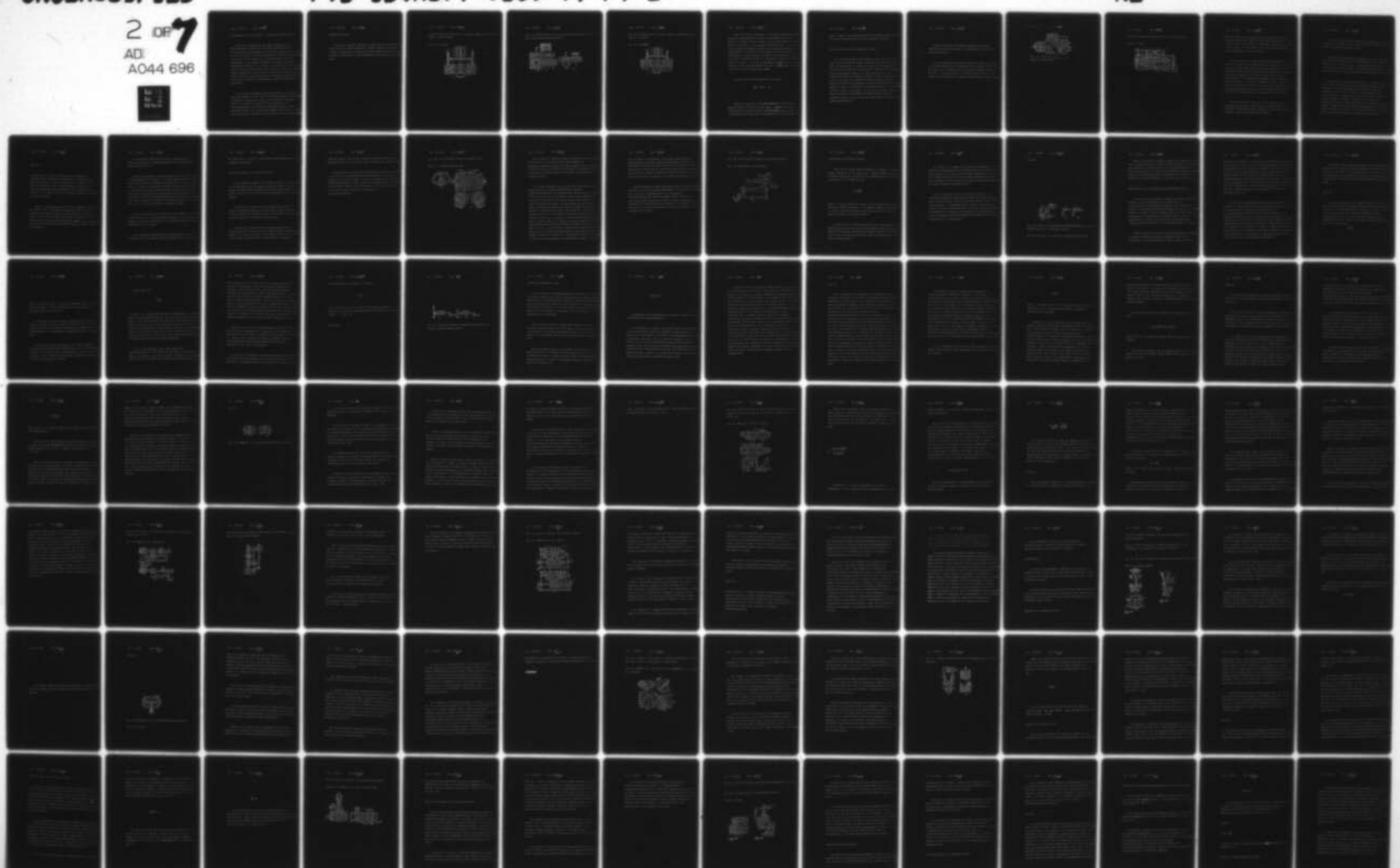

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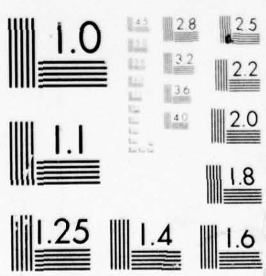

MICROCOPY RESOLUTION TEST CHART
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electromagnet 1, but the second - in the form of sleeve valve 4.

During the supplying into the armature winding of the electromechanical converter of electrical control signal, it deflect/diverts shutter/valve 2 of the neutral position to one side or the other. With the aid of shutter/valve in working chambers b and a of valve 4 of the square of the intensification, held in the mid-position by the centering springs, is created the pressure differential, utilized for a control of the plunger of this valve. In these devices the valve ensures fluid flow rate, applied to hydraulic engine, proportional to the force of differential electrical input current. After the valve will engage the position, proportional to the assigned deviation of shutter/valve, appears the feedback on effort/force.

By the design of feedback, they distinguish as in previously described systems (see Fig. 115a), also servo hydraulic boosters with lever/crank feedback (Fig. 135a), the coefficient of feedback of which it is possible to change with the selection of the relationship/ratio of the arms of differential lever, and also hydraulic boosters with rigid single feedback (Fig. 135b), in which the sensing device (nozzle) is placed directly on exit component/link

(distribution valve).

Besides the examined distributors, which ensure at output/yield fluid flow rate, proportional with a constant pressure differential to input displacement, are applied the distributors, which ensure the pressure, proportional to input displacement with constant fluid flow rate.

Fig. 134. Hydraulic system with a two-stage amplifier of the type" nozzle" - shutter/valve.

Key: (1). To engine.

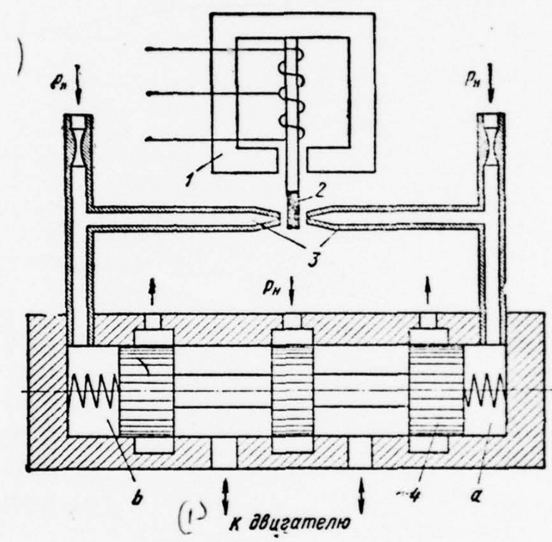


Fig. 135. Hydraulic boosters: a) with lever/crank feedback; b) with rigid single communication/connection.

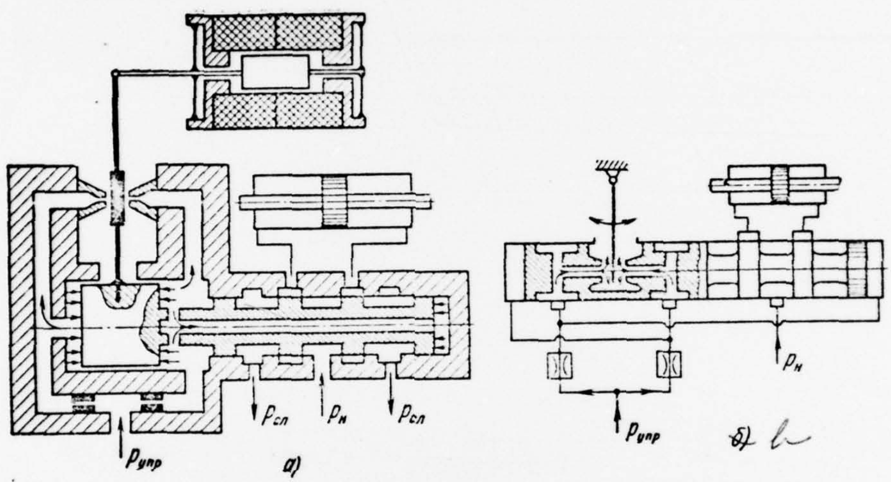


Fig. 136. A distributor of the type" nozzle" - shutter/valve with control on pressure.

Key: (1). To engine.

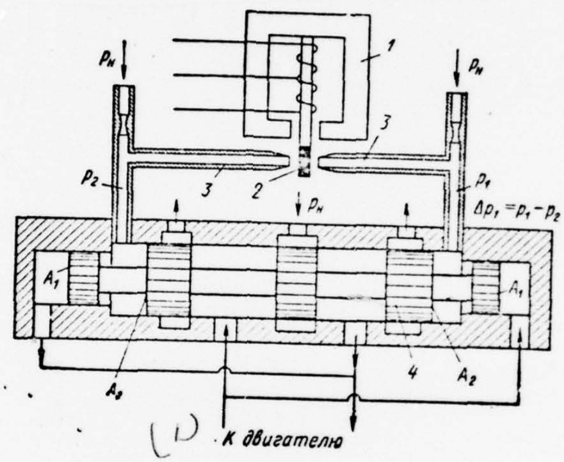


Figure 136 shows the schematic of the similar distributor which ensures at output/yield the pressure, proportional to the force applied to entrance electric current. By the output signal of first stage of this device is the pressure differential $\Delta p_1 = p_1 - p_2$ in the interthrottle chambers (nozzles) 3, created the displacement of shutter/valve 2 relative to the section/shear of these nozzles, proportional to current strength, applied to electromagnet 1. This the pressure differential acts on a difference in areas $A_2 - A_1$ the plunger of valve 4. On surface A_1 the band of a small section acts a jump/drop in the operating pressure of Δp_p, proportional to working lcad on the piston of hydrostatic engine.

Condition of the static equilibrium of plunger

$$A_1 \Delta p_p = \Delta p_1 (A_2 - A_1).$$

During the supplying to the electromagnet of 1 differential electric current, the force of the $\Delta p_p A_1$, created the pressure differential in hydraulic engine, not equilibrant $\Delta p_1 (A_2 - A_1)$, by the created first step/stage of intensification, and the plunger of

valve is moved to the appropriate side to the value, proportional to the created difference in the forces of electric currents.

Servo hydraulic drive with volumetric control.

By a deficiency/lack in the systems of choke control is the power loss with flow-rate control throttling. In view of this they apply, and especially in systems with large (8-10 kW) powers, the servo hydraulic drive with the volumetric control of exit velocity, realize/accomplished change in the pump capacity (Fig. 137). In similar drive with the volumetric control in which are applied the pumps of the adjustable expenditure, occurs feed mode with as variable pressure and expenditure. Since the amount of working fluid, applied by pump, is determined by the required velocity of performing hydraulic engine, and pressure - by its load, the source power of supply corresponds not allowing for power losses, consumed by hydraulic engine. In view of this this drive differs in terms of high energy characteristics.

Figure 137 depicts the schematic diagram of the similar servomechanism with radial pump 1 and servomotor in the form of actuating cylinder 5.

The feed control of pump and the reverse of feed in this drive is realize/accomplished by displacement of its housing relative to the fixed axis of cylinder block 2, performed through thrust/rod 9 and lever 8, one side of which is connected with stock/rod 3, but another - with thrust/rod 7, connected with entrance (control knob).

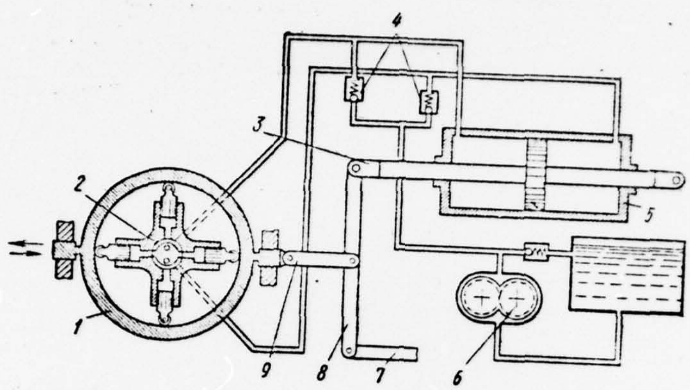
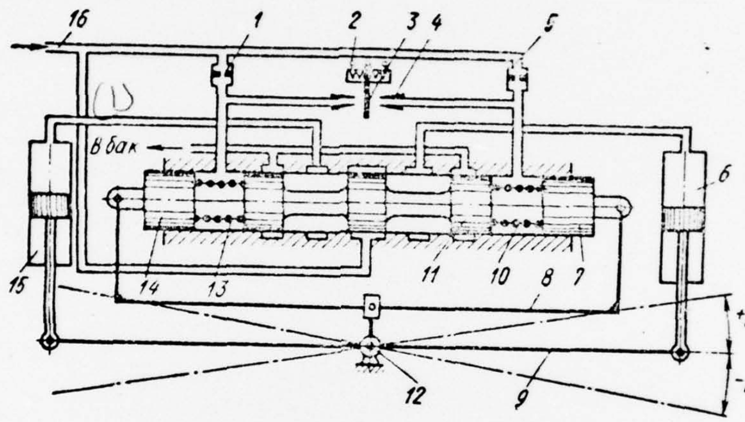


Fig. 137. Hydraulic booster with adjustable feed pump.

Fig. 138. Diagram of two-stage control of the feed of pump.

Key: (1). In tank.



System is supplied by the auxiliary gear pump 6 pumping, which supplies through check valves 4 liquid under pressure 35-40 kgf/cm² to pump 1 and to both cavities of cylinder 5, maintaining pressures in the cavities of this cylinder at the zero feed of adjustable pump 1 and fixing with the fact its piston.

For a control of the pump of alternating/variable feed in the systems of automation, are applied the electrohydraulic amplifiers, in which as input effect serves electrical signal. This signal, intensified by electronic (or another type) amplifier, is converted by the electromechanical converter of steering gear into the displacement of the controlling cell/element of the auxiliary hydraulic booster of the control mechanism of pump. In turn,, the control signal, which enters the entrance of this hydraulic booster, is amplified according to power up to the value, necessary for a control of the organ/control of a change in the feed of pump.

Thus, in dependence from value and the polarity of the electrical error signal at the entrance of servo system can be realized the infinitely variable control of the exit velocity of hydraulic engine and connected with it load.

Such drives possess high sensitivity and stability, in them it is possible to utilize for the formation of the serrated signal of control a system of electrical sensors.

Analysis shows that automatic systems in the very best manner satisfy the high-speed position electrohydraulic servo systems in which the feed control of pump is realize/accomplished with the aid of steering gear, which consists of low-power electromagnet and the two-stage positioned pozitsionirovannogo auxiliary hydraulc actuator.

Specifically, are common the schematics of steering gears of the feed of pump, which have as mechanical converter position type reversive rotary electromagnet, which affects the two-stage distributor of the type the" nozzle" - shutter/valve with the included feedback by valve, which through the special hydraulic booster converts the angle of rotation of the cylinder of electromagnet into the angle of rotation of the controlling cradle (slant) of pump. Figure 138 depicts the schematic of the two-stage hydraulic booster of control of the feed of the axial-reciprocating pump, realize/accomplished by the rotation of its cradle.

Page 186.

The entrance of the first cascade/stage is controlled by electromechanical converter 2 shutter/valve 3, arrange/located between two nozzles 4, and by output/yield - the plunger of 11 distribution valves, that is simultaneously the entrance of the second cascade/stage appear cylinders 6 and 15 mechanism of the rotation of the cradle of pump, that turn its relative to axle/axis 12 through angle $\pm\gamma$.

Plunger 11 is stopped up with the aid of springs by 10 and 13 two pistons 7 and 14, connected through rigid frame 8, which is connected with rocking yoke/arm 9, strengthened on the axis 12 of turning of the cradle of pump. Rigid frame 8 and pistons 7 and 14 with springs 10 and 13 realize/accomplish feedback of the transmitter of the second cascade/stage of hydraulic booster (cradle of pump) with plunger 11.

For operational provisions of nozzle - shutter/valve in hydraulic system, are established/installed two throttle/choke of fixed resistor 1 and 5.

During the supplying of electrical signal to the control winding of electromechanical converter 2, its rotary armature together with shutter/valve 3 will be deflected to certain angle to that or other-side, as a result in the cavities between springs 10 and 13 valves it is created the pressure differential, which will move plunger 11, cause by the displacement of shutter/valve, will be compensated for, and plunger 11 will be establish/installed at free position. In this case, it will overlap the window of the supply of actuating cylinders 6 and 15, as a result the cradle of pump will be fixed.

Thus, by changing value and polarity of the governing electrical signal, applied into electromechanical converter 2, is represented possible to ensure the deflection of the cradle of pump of the angle, proportional to this signal.

By the important advantage of the examined devices is the possibility of the velocity control of hydraulic engine by means of

the direct feed of signal to electromagnet from the potentiometers of automatic controllers.

Hydraulic boosters of the torsional moment.

By the hydraulic booster of the torsional moment, is understood the servomechanism with the hydraulic engine of rotary or rotary action, connected with distributor (error meter) which usually is carried out in the form of rotary valve (tap/crane) with the tracking bushing.

In practice are common rotary type amplifiers in which are applied the hydraulic motors of rotary action. Similar hydraulic boosters convert by torque/moment into the synchronous rotary motion of output/yield with the higher torsional moment.

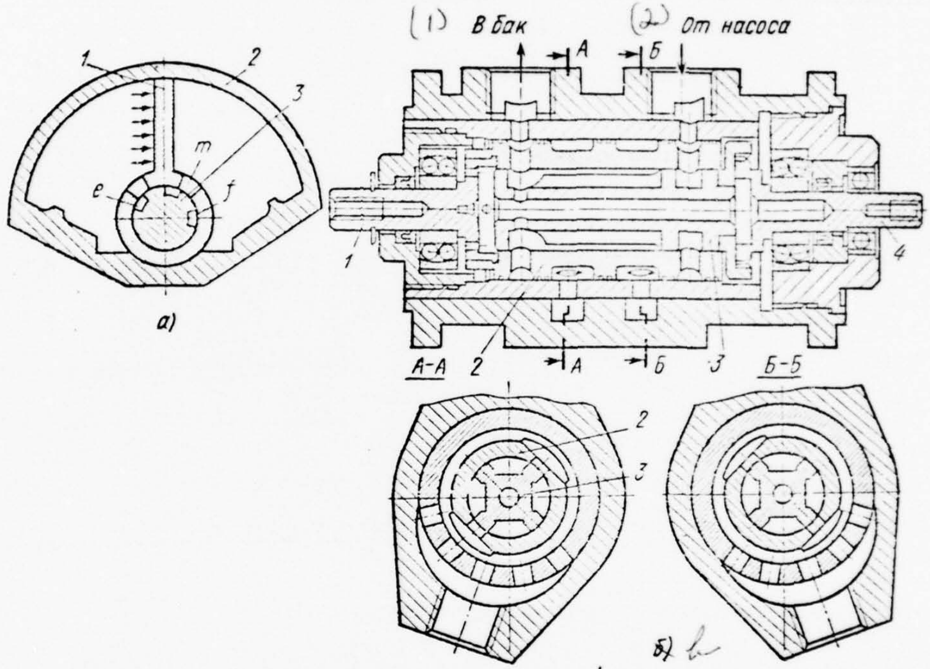
The amplifier circuit of rotary type torsional moment is represented in Fig. 139a. Piston has a form of plate (blade/vane) 1, which can be turned in housing 2 at an angle of <360°. The butt end of this plate, connected with the output shaft of performing

hydraulic engine, serves as its axle/axis, and also the bushing of distribution valve 3 of the rotary type, connected with control knob.

Liquid under pressure will be fed through the channels of valve e and f, but it is abstract/removed through channel m. During the rotation of the plug of valve 3 relative to bushings, will occur the disagreement/mismatch of their positions, and liquid, after entering the appropriate cavity, it will turn plate 1 in the same direction, as the plug of valve, before the elimination of disagreement/mismatch.

Fig. 139. Servo hydraulic boosters of rotary action.

Key: (1). In tank. (2). From pump.



Valve 3 with the tracking bushing it performs in this circuit of the function of the sensing device, which measures the disagreement/mismatch of input and output shafts. Since the bushing of valve is strictly connected with the shaft of hydraulic engine, system valve - hydraulic engine is enveloped by reverse/inverse stiffening joint.

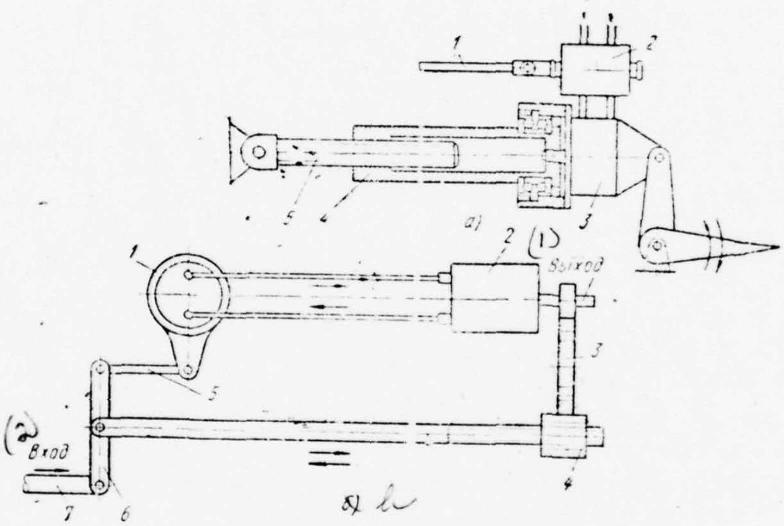
In the servo hydraulic drive of full-turn rotary motion (with hydraulic motor) and with reverse/inverse mechanical communication/connection are commonly used the distributors (valves) of the rotary type with the tracking freely sponsor bushing which fulfills the functions of the sensing device, which measures the disagreement/mismatch between the input signal and the feedback signal, whereupon distributor is carried out usually in the form of independent apparatus (Fig. 139b). The bushing of 2 distributors is connected with the aid of kinematic vapor with the output shaft of hydraulic motor, i.e., to bushing 2, enters feedback signal, and to valve 3 control signal. As a result the bushing, connected through feedback with hydraulic motor, tracks the motion of the plug of valve. For this, the distributor is installed in system so that the crank (lever) of control is linked beaded 4 plugs, and the shaft of feedback - beaded of 1 bushing of 2 distributors. During the rotation of plug 3, are open/disclosed the corresponding channels, employed

for the supply of the hydraulic engine whose rotation through reverse/inverse mechanical communication/connection (reducer) is transferred to bushing 2, which is turned to the same side to which was turned the plug. With the cessation of plug (with the cessation of the motion of governing knob/stick) bushing 2 attacks to it and overlaps the channels (windows) of the supply of hydraulic engine.

Plug and bushing in steady conditions rotate at constant angular velocity with the error angle, proportional to the velocity of motion and to the load of hydraulic motor (it corresponds to discovery/opening the throttling window). With an increase in velocity and fluid flow rate, the area of windows increases, and consequently, displacement angle increases. For a decrease in the dead zone, the overlap by the plug of channels in bushing 2 must be minimum (0.05-0.08 mm).

Fig. 140. Servo hydraulic boosters with hydraulic motors.

Key: (1). Output/yield. (2). Entrance.



Self-braking (irreversive) systems.

The filled by liquid actuating cylinder is similar to hydraulic spring. Expression for the hardness of a similar hydraulic spring of actuating cylinder not allowing for the deformation of the walls of cylinder (see p. 19) takes the form

$$C_{z}=\frac{2ES^{2}}{V},$$

where E - the bulk modulus of liquid; S is the effective area of the piston of actuating cylinder; V - the working volume of the chamber of cylinder (hydraulic engine) and of main line, that is located under the operating pressure of liquid.

In view of this if necessary provision for a high hardness and nonreversing nature of the system of rectilinear forward motion, are applied the high-speed hydraulic motors whose rotation is converted with the aid of mechanical node/units (worm and helical vapor) into exit rectilinear motion.

The hardness of a similar transmission in comparison with the hardness of transmissions with actuating cylinder is caused by the small volume of the liquid, which undergoes compression. So, if in the actuating cylinder of rectilinear motion compression undergoes only the volume of liquid, which is located at this torque/moment in the working chambers of hydraulic motor.

Figure 140a depicts one of the possible circuits of the self-braking hydraulic transmission of throttle control. Hydraulic motor 3 rotates bushing 4 with the internal cutting in which enters screw/propeller 5. Distribution valve 2 connect with thrust/rod 1 system of feedback with control knob. The static nonreversing nature of transmission is ensured by the selection of the corresponding angle of screw/propeller and kinematic characteristics of the other component/links of transmission.

Page 189.

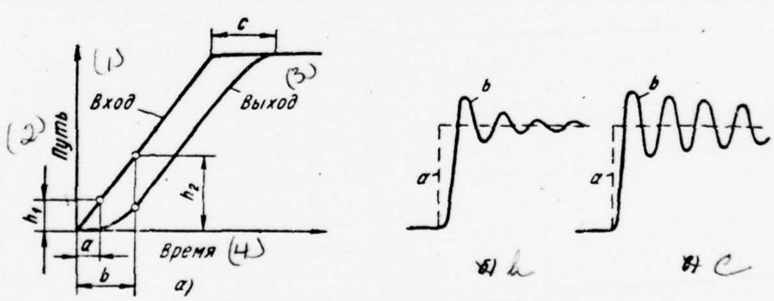


Fig. 141. Curves, that characterize accuracy/precision (a) and the stability (b and c) of hydraulic booster.

Key: (1). Entrance. (2). Way. (3). Output/yield. (4). Time.

Figure 140b shows the schematic diagram of the hydraulic booster of the torsional moment, in which the velocity control and reversing the output/yield of hydraulic motor 2 are realize/accomplished by a feed control of pump 1. For this, the thrust/rod 5 control of pump is connected by differential lever 6 with control rod (entrance) 7. Feedback is realize/accomplished by gears by 3 and screw pairs 4.

SENSITIVITY AND THE ACCURACY/PRECISION OF HYDRAULIC BOOSTER.

one of the basic requirements, imposed to servo type hydraulic boosters, is the requirement for the provision for an accuracy/precision and sensitivity hearth by which understands the complex of the qualities, which characterize the ability of hydraulic booster to reproduce with the minimum error (on time and way) the displacement of output/yield in accordance with the assigned displacement of entrance, the error over time characterizing operating speed, but by way of - the accuracy/precision of system.

Figure 141a gives the traffic, which characterizes the quality of hydraulic booster in question. Curve/graph shows that the displacement of entrance (plunger of valve) in way h_1 from free

position (it corresponds to the interval of time from the beginning of the motion of entrance a) it is not accompanied by the motion of output/yield (piston of actuating cylinder). This way characterizes the dead zone of system. During the further displacement of the entrance, starts up the output/yield; however, its velocity is establish/installed only after passage by the entrance of certain way h_2 (it corresponds to the interval of time b). In equal measure the motion of output/yield at the end of power stroke will cease itself only after certain time after the cessation of the entrance, characterized by cut c.

The examined disagreement/mismatch in the displacements of entrance and output/yield (path error) determines the error (error) of tracking, which characterizes sensitivity and the accuracy/precision of servo system. This error depends on the high-speed/velocity and power operating mode of hydraulic booster. Specifically, the error in the tracking in steady conditions with the loaded output/yield usually is determined by the distance (way) h_2, up to which must be displaced the plunger of valve (entrance) from free position at the position, which ensures pressure and fluid flow rate, required for the overcoming of load and development of the given speed of the steady motion of output/yield.

In transient (being unsteady) condition can occur an additional increase in the error (disturbance/breakdown of accuracy/precision) because of the action of the pressures exerted by masses, connected to output/yield, for overcoming of which will be required larger than during steady conditions of motion, discovery/opening slide valve ports.

Page 190.

The error is determined by a number of factors, also, first of all by the gear ratio of kinematic feedback loop, which characterizes the ratio of the displacement of the plunger of 3 valves (entrance) to the displacement of the piston of 4 hydraulic engines (output/yield) with of the fixed control rod with motionless output/yield (see Fig. 115). For the examined circuits, depicted on Fig. 115, this gear ratio

$$i=\frac{m}{n},$$

where m is reach of lever 7 between the attachment points of valve and control rod; n is the same between the attachment points of hydraulic engine and control rod.

In other words, for the circuit, depicted on Fig. 115a, this number is always less than 1, but for a circuit in Fig. 115b - is always more than 1. This number frequently they assume/take, if it is allow/assumed by the requirements for stability, equal to five and above.

For the servo system, presented in Fig. 115b, is possible principally how conveniently to decrease this error at output/yield by means of an increase in the intensification in speed (by means of increase $i = m/n$). However, this increase practically is limited by the stability conditions of system.

Besides gear ratio

$$i=\frac{m}{n}$$

for error in the tracking they influence the airtightness of system, gaps in its mechanical node/units, loads and on exit velocities and a series of the other factors, which affect dead zone. Separately one should note static friction (the frictional rest) in the nodes of system, and in particular in valve, which it can lead to considerable delay in the reaction of controls to control signals. Under known conditions the tracking input signal at low speeds will be stepped, and work of servo system will become intermittent.

Not the less important factor, which affects the accuracy/precision of servo system, are the gaps in the kinematic chain of the systems, as a result of which is reduced its sensitivity and increases "dead space". In the presence of gaps, the motion of

the output/yield of the system at first of their selection occurs without load, whereupon, thus far the leading component/link (entrance) does not pass the way, equal to gap, driven/known component/link (output/yield) remains motionless and will initiate to be moved only after in the leading component/link will be selected gap. The leading component/link it acquires for this time interval certain speed, and only after this both component/links will be moved jointly. Analogously during the motion of the leading component/link to the side of the initial position driven/known component/link will initiate to be moved as in the first case, only after passage by the leading component/link of certain way, which corresponds to gap.

Analogous effect on the accuracy/precision of system exerts the elasticity of its components. It is obvious that the displacement of output/yield after the start of entrance will be initiated only after the voltages of all elastic cell/elements, including liquid and conduit/manifolds, they will achieve/reach the value, which corresponds to the load of output/yield.

In practice the evaluation of accuracy/precision and sensitivity of hydraulic booster frequently produces itself according to the ratio of the establish/installed exit velocity of the aaaa of piston

to displacement χ the plunger of the valve:

$$k = \frac{v_n}{\chi},$$

which accepted to call the quality of servo system (hydraulic booster). Depending on designation/purpose and quality of hydraulic booster k = 10-125 /s

end section.

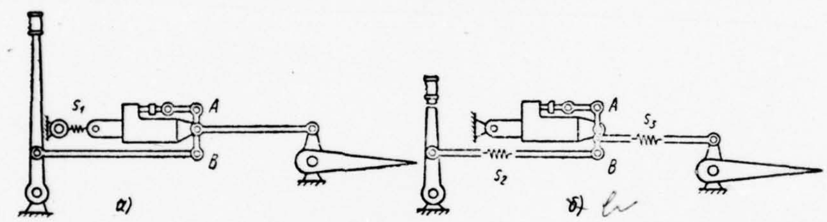


Fig. 142. The design diagrams of hydraulic boosters with the connected elastic component/links.

STABILITY OF HYDRAULIC BOOSTER.

By the stability of hydraulic booster, is understood its ability to return to steady state after the break-down of the source, which destroyed this equilibrium. Figure 141b and c gives the curve/graphs, which characterize the stability of two systems one of which has high (Fig. 141b), and another - low stability (Fig. 141c). The curve of a expresses the displacement of entrance, while the curve of b - the displacement of output/yield.

Steady conditions can be disrupted both change in the input parameter (reference input) and by the change in the exit parameter (for example by the external perturbation effect in the form of a change in the aerodynamic loading on the aircraft control surfaces, etc.).

Stability largely affects the kinematics of the actuator of distributor. Thus, for instance, with an increase in the gear ratio of differential lever 7 feedback (see Fig. 115b) the probability of destabilization it is raised. The acceptable from this viewpoint gear ratio of this lever it is

$$i = \frac{m}{n} = 3 \div 4.$$

Consequently, the requirements for stability are opposite to the requirements for accuracy/precision.

On the stability of system, great effect exerts the elasticity of conduit/manifolds, liquid and mechanical component/links of system, and in particular the elasticity of communication/connections of hydraulic engine with load, and also the elasticity of those machine parts to which are fastened the power packs of hydraulic booster. During some external and internal disturbance/perturbations and during their combination with the mass load, elasticity and the design features of system, it can enter the resonance oscillations during which can develop considerable dynamic loads.

Figure 142a shows the diagram of elastic fixing to the airplane of the stock/rod of the cylinder, which belongs to the hydraulic booster of the system of controls, equipped with sensitive (with low coverage) valve. It is not difficult to see that in the presence of this elasticity the driving pulse, which causes the bias of the exit component/link (actuating cylinder) of hydraulic booster, it can cause the oscillation of system. With this momentum/impulse/pulse the attachment point of the stock/rod of cylinder as a result of elasticity will be displaced and will turn (with motionless knob/stick) the lever of feedback AB, after displacing valve. It is obvious that with the known elasticity s_1 the plunger of valve will be able to be displaced so, that the liquid will enter the appropriate cavity of actuating cylinder and will set it in the motion during which will change the direction of the motion of lever AB. Accumulation in this case elastic energy will contribute to the passage of the valve through the equilibrium (neutral) position. If valve possesses high sensitivity, then the described process will be repeated in opposite direction, as a result of which they can arise auto-oscillation with the natural vibration frequency of exit component/link.

Page 192.

Figure 142b depicts the diagram of the reversible hydraulic booster, in which part of the acting at output/yield effort/force is transferred to control knob. Let us assume that the attachment of actuating cylinder absolutely rigid, and input and output circuits respectively they possess elasticity s_2 and s_3. The destabilizations of system in this case it can be caused by sharp disturbance/perturbations both from the side input and output circuits. Thus, for instance let us assume that output/yield under the action of any external power momentum/impulse/pulse (action of air flow to control plane) it will be displaced toward one side or the other. The rotation of the control through the output circuit and the differential lever AB is transferred to the input circuit, the elastic component/link s_2 which it will make possible to lever AB to turn itself and to discover distribution valve, which, in turn, will cause the displacement of actuating cylinder. As a result with the known relationship/ratios of the mass load and hardness input s_2 and exit s_3 ducts the energy of liquid will cause the self-excitation of system. In this case, elasticity s_2 on input circuit will lower the attenuating effect of the mass of knob/stick and control rods and decrease the stabilizing effect of the action of operator on control knob.

Compressibility effect of liquid and deformation of conduit/manifolds. Effect on the stability of the system of the compressibility of its filling liquid and the elastic deformation of the walls of hydroaggregates and conduit/manifolds is analogous with the effect of the elasticity of the mechanical nodes of hydraulic system examined above. Under this condition the liquid, which enters from pump, is compressed during part of the oscillatory cycle, and also expands conduit/manifolds, accumulating energy. In the subsequent part of the oscillatory cycle the accumulated in liquid and conduit/manifolds, energy, entering system, is added to the energy, which comes in from pump. The hydraulic cylinder, filled by liquid under certain pressure with the overlapped channels of entrance and output/yield, can be likened in this case to the spring whose hardness will be determined by the compressibility of liquid.

It is not difficult to see that the linear elasticity (sag) of a similar liquid spring is equal (deformation of the walls of cylinder we disregard)

$$C=\frac{\beta'}{4A^2},$$

where β - the coefficient of the compressibility of liquid; V is a total volume of the closed (blocked) liquid in cylinder; A - effective piston clearance.

Analogous sag will be observed, also, in diagram with the hydraulic engine of rotary motion (hydraulic motor). Let us assume which valve airtightly overlaps both the channel of the filled by liquid hydraulic motor, connected with input and exit conduit/manifolds. If liquid was incompressible, then the shaft of hydraulic motor would be strictly stopped up by the closed liquid (by deformation of the part of hydraulic motor and nonhermetic state we disregard). However, since liquid is compressed, the shaft of hydraulic motor can be turned to any angle. In this case, the hydraulic motor acts as pump, raising pressure in one cavity and after reducing in another. The torque/moment, necessary for the rotation of the shaft of hydraulic motor, is proportional the

pressure differential in conduit/manifolds a angle of rotation - to a change of the volume of the cavities of hydraulic motor, connected with conduit/manifolds. Assuming that the bulk modulus of liquid appears for the acting pressures by constant, it is possible to consider that the shaft of hydraulic motor connect with the spring of permanent hardness.

The angular elasticity of equivalent spring is equal for this case

$$C_1 = \frac{\beta V}{4q_1^2} = \frac{\pi^2 \beta V}{q_2^2} \text{ rad/ (cm•kg)},$$

where q_1 and q_2 - the specific working volume of hydraulic motor in cm^3/rad.

In this case, we assume that the closed volumes of liquid in the opposite cavities of hydraulic engine (gutter and forcing) are equal between, by itself.

Page 193.

In connection with actuating cylinder this condition will be observed for the mid-position of piston with bilateral stock/rod (see Fig. 26b).

From the point of view in question especially is undesirable the presence in the liquid of air. Presence in the cylinder of the hydraulic booster of the undissolved air gives as a result of compression to the storage of kinetic energy during the oscillations of stock/rod (output/yield) with its subsequent return to motion.

It is obvious that in the slow-operating servo systems the effect on the operating mode of the elasticity of liquid usually is negligibly small. However, in the case of inertia loads with considerable static friction (for example in the case of the drive of the table of planer) the effect of elasticity can lead to the intermittent motion of the output/yield of drive. This is caused by the fact that through the valve must pass certain amount of liquid before in the cavities of hydraulic engine will be created the pressure differential, sufficient for the overcoming of the

frictional forces of rest and inertia. After this for motion output/yield with load, are required because of a reduction in friction less forces, as a result will occur the excess/throw/overshoot of the velocity of motion. After the cessation which will occur practically at the adjusted pressures in the cavity of engine, appears static friction, whereupon cycle will be repeated.

It is obvious that the examined effect of springing is aggravated during the application/use of a liquid with low elastic modulus and with the large volumes of the working cavities of hydraulic engine. For this reason frequently (specifically, in heavy machine tools, in control of the aircraft control surfaces, etc.) it is necessary to reject the application/use of power cylinders, replacing them by the hydraulic motors, the volume of cavities of which is considerably less than the volume of the first.

For hydraulic motors important, from the viewpoint of the effect in question, is the volume of the dead space of the motor hearth by which is understood a difference in the maximum volume of its chambers (with coupling channels) Vi of working volume q. This parameter is characterized by coefficient

$$k_v = \frac{q}{V - q},$$

where q and V - q - working volume and the volume of the dead space of hydraulic motor.

Better/best from this viewpoint axial-piston hydraulic motors even when the coupling channels are carried out shortest possible and a small section, have an $k_v = 4$, for the series machines of $k_v = 6 \div 10$.

Effect of gaps. The source of the onset of oscillations can be also gaps in the component/links, included in feedback (gaps in the chain/network of articulation from control knob to distribution valve), which amplify the oscillations, caused by the elasticity of lever/crank system and thrust/rods, during appendix to them of load or are the source of the onset of oscillations. Specifically, in the presence of gaps can occur the loss of stability as a result of the

axial lack of balance of valve, produced by the hydrodynamic action of fluid flow. It is obvious, if on the input circuit of system is a gap, then the plunger of slide valve under the action of the oscillating hydrodynamic force will be displaced on the limits of this gap, which can with the high sensitivity of valve cause the flow reversals of liquid and oscillation of system.

Methods of an increase in the stability. The simplest method of an increase in the stability of hydraulic booster is an increase in the overlap of valve (increase in the dead zone) and a decrease in the intensification of system in velocity. However, a similar method makes operating speed and the accuracy/precision of hydraulic booster worse. From that which was presented it follows that an increase in sensitivity and accuracy/precision of system is achieved by the application/use of the valve with the minimum overlap, which moreover must ensure with slow speed sufficiently large passage cross sections for a liquid, whereas for an increase in the stability of system against oscillations overlap one should increase, and the flow areas decrease.

Page 194.

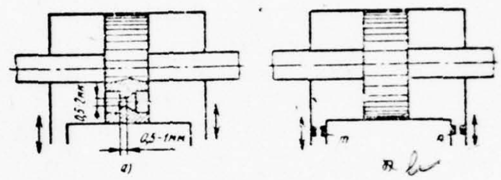


Fig. 143. Schematics of the arrangement/permutation of dampers.

An excessive increase in the overlap can cause as a result of an excessive increase in the dead zone inverse effect - fall in the stability.

The factor, which stabilizes system, is its nonhermetic state; therefore for an increase in the stability, frequently are introduced the artificial leakages, which scatter during oscillations part of the energy of system. The attenuating properties of leakages are caused by energy loss, and also fact that in the presence their increases the dead zone.

Artificial hydraulic slip in cylinder usually creates by shunting through the resistance (throttle/choke) of the cavities of cylinder, for which in its piston, usually it is carried out opening/aperture 0.5-2 mm in diameter (Fig. 143a).

Stabilizing effect exerts also line resistance and the local losses. For this, at entrance or output/yield of liquid from actuating cylinder or simultaneously in entrance and exit are establish/installed throttle/chokes m and n (Fig. 143b).

Attenuation of vibrational energy. The most radical method of the extinguishing of oscillations is the application/use of dampers of the hydraulic resistance with the aid of which the vibrational energy is scattered in the form of heat.

Damper is establish/installed in the oscillatory system between any two by that being moved one relative to another by points (usually between the being moved part of distributor and its housing). In systems with slide-valve distributor damper usually is linked with plunger, and the bushing of damper - with distributor housing.

Damper is cylinder small piston 3 which (Fig. 144a) is connected with the plunger of 1 valve. in piston is located throttle/choke 2. During the displacements of small piston 3, liquid is displaced by it through this opening/aperture of one cavity of cylinder into another. For the elimination of the overflowing of the liquid through the radial slot between the small piston and the cylinder applied sealing ferrule 4. In the absence of this ring, certain part of the liquid would flow/last in work of the damper through the radial slot, since

the amount of overflow liquid depends on the alternating/variable size/dimensions of slot and viscosity (temperature) of liquid, a change in these factors would disrupt the operational stability of damper.

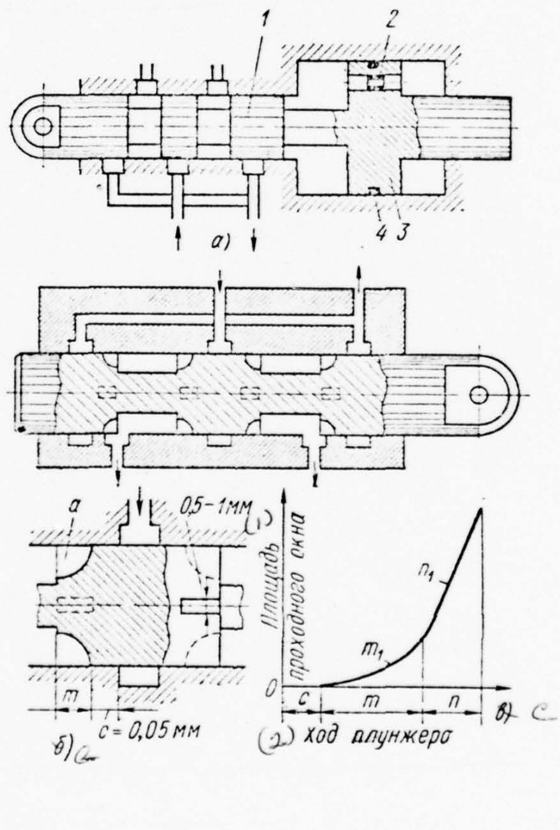
The section of throttle/choke 2 in the small piston of 3 dampers is selected so that attenuation, on one hand, would not increase the effort/forces, necessary for the displacements of valve during control a on the other hand, in order that during its high-speed/velocity oscillatory displacements would be created the attenuation, capable of extinguishing the force, exciting oscillations. So, with the diameter of dash-pot piston 30-40 mm and the oil of AMG-10 the diameter of throttle/choke 2 usually is 0.6-0.8 mm.

The calculation of hydraulic damper is reduced to the determination of flow resistance of the liquid, forced by the small piston of the damper through the choke opening/aperture, performed usually taking into account flow of liquid through opening/apertures in fine/thin wall. The pressure differential Δp in the cavities of the cylinder of damper, created by the resistance of throttle/choke, and expenditure Q liquids are connected in this case by dependence

(20). Coefficient of the expenditure for a round opening/aperture in the fine/thin wall of piston $\mu = 0.62$.

Fig. 144. Valve with damper (a) and the stepped section of passage openings (b) and the dependence of the area of window on piston stroke (c).

Key: (1). Area. (2). Course is plunger.



Taking into consideration that the indicated expenditure is equal, under the condition of the absence of hydraulic slipes through the clearance between the small piston and the cylinder, to the volume, described by dash-pot piston Q = vF, where v and F - the speed of displacement and the piston clearance of damper, we can write

$$Q=Fv=\mu f\sqrt{\frac{\Delta p\cdot 2}{\rho}}$$

or

$$\Delta p=\frac{\rho}{2\mu^2}\cdot\frac{F^2v^2}{f^2}.$$

Parameters F, v f and Δp are selected so, in order to effort/force P = FΔp, necessary for the displacement of the small

piston of damper at its required maximum working speed, it would not exceed rated values.

When damper does not have ferrule 4 (see Fig. 144a), one should consider the overflowing (expenditure) of the liquid through radial clearance (slot) between the small piston and the cylinder. The calculation of damper in this case is recommended to carry out in the following order. First one should calculate according to expression (23) for an eccentric slot the fluid flow rate of $Q_{щ}$ with the assigned pressure differential Δp through the radial slot. Deducting this expenditure from volume Q = vF, described by dash-pot piston at its assigned speed is determined the volume, which must be extrude/excluded for the same time with the assigned pressure differential Δp through the choke opening/aperture in the piston:

$$Q_1 = Q - Q_{щ} = vF - Q_{щ}.$$

With this expenditure the opening/aperture for providing the required pressure differential must have an area according to expression (20)

$$f=\frac{Q_1}{\mu\sqrt{\frac{\Delta p\cdot 2}{\rho}}}=\frac{vF-Q_{щ}}{\mu\sqrt{\frac{\Delta p\cdot 2}{\rho}}}.$$

When the fluid flow rate of the $Q_{щ}$ through the radial slot with the assigned pressure differential, calculated according to expression (20), will be more than Q = vF, i.e., when the damper does not ensure the assigned resistance even without supplementary opening/aperture in small piston, necessary either to decrease the radial clearance in piston vapor of damper or to increase the diameter of its small piston.

Page 196.

Since the volume of damper of V = LF vary directly the square of its diameter, while the perimeter of radial slot (length of the

average circumference of slot) increases in proportion to the first degree of this diameter, and the pressure differential under otherwise equal conditions vary indirectly the square of the diameter of small piston, by a change in this diameter it is possible to regulate the relationship/ratio of the operational characteristics of damper. From the viewpoint of reliability and stability of attenuation, it is expedient to maximally increase area F of small piston with a simultaneous decrease in the section of opening/aperture f of the throttle/choke, minimum diameter of which, however, must be not less than 0.3 mm.

The energy, absorbed during the extrusion of the liquid through throttle/chokes for one piston stroke (not allowing for friction of piston in cylinder),

$$E_0 = LF\Delta p,$$

where L is the travel of the piston of damper (range of oscillation of valve).

Application/use of valves with stepped passage openings. For an increase in the stability of hydraulic booster, are applied the valves with stepped in the course of plunger by the section of

passage openings (Fig. 144b), whereupon with the first part of it way of window they are made with variable resistor. This is reached by the fact that supply of liquid in the beginning of the motion of plunger realize/accomplish not on entire perimeter of the circumference of the plunger of valve, but through narrow grooves (slot) a of alternating/variable depth. Practically flow area into parts of length m of the course of valve (approximately at length 0.2-0.5 mm from neutral line) make in the form of two or four evenly arranged/located through circumference narrow arc slots, which have alternating/variable section.

The curve/graph of a change in the section of slot in the course, the showing fracture of curved system performance, are represented in Fig. 144c. In the first after passage dead space c of part of the piston stroke an increase in the section of slot at length m is characterized by the curve m_1, an increase in the section at length n is the curve of n_1.

Stabilization of system by the improvement of feedback. The method of an increase in the transient stability (stabilization) of drive described above by hydraulic slip is connected with its essential supplementary expenditure, and an increase in the transient

stability by attenuation is connected with a fall in the operating speed of system.

Therefore frequently is applied stabilization by the improvement of feedback, and in particular by the application/use of a flexible feedback. By means of the rational construction of feedback is represented possible to improve the quality of transient process (to accelerate fading transient process and to decrease its duration), and also to decrease the static error.

Figure 145a gives the schematic of two-stage hydraulic booster, in which for an increase in the stability proportional control of mismatching is corrected by control on first-order derivative (on the rate of change in the disagreement/mismatch). The basic distribution valve of 1 this schematic, which feeds actuating cylinder of 9 hydraulic booster, is driven by the piston of auxiliary actuating cylinder 2, controlled by auxiliary valve 8, connected with differential lever 5 feedback with entrance (control knob).

The correction of proportional control is realize/accomplished as follows. The piston rod of auxiliary cylinder 2, connected with

the plunger of basic valve 1, will bear on himself the piston of 7 cylinder of 6 damper which, in turn, is connected with differential lever 5 feedback, but it has the capability of axial displacement, from which it is held by spring 4. The displacement of entrance (auxiliary valve 8) causes the displacement of the piston of auxiliary cylinder 2, and together with it and the piston of 7 cylinder of 6 damper. However, the displacement of piston 7 in the cylinder of damper is possible only after the displacement through the throttle/choke 3 liquids from its appropriate cavity. In view of this during sharp displacement piston 7, for example, to the right cylinder 6 into the first instant is moved as a result of the high resistance of throttle/choke together with this piston, compressing spring 4 and changing with the aid of lever 5 discovery/opening slide valve ports 8.

Fig. 145. Schematics of hydraulic boosters with the correction of proportional control.

Key: (1). Entrance. (2). Output/yield.

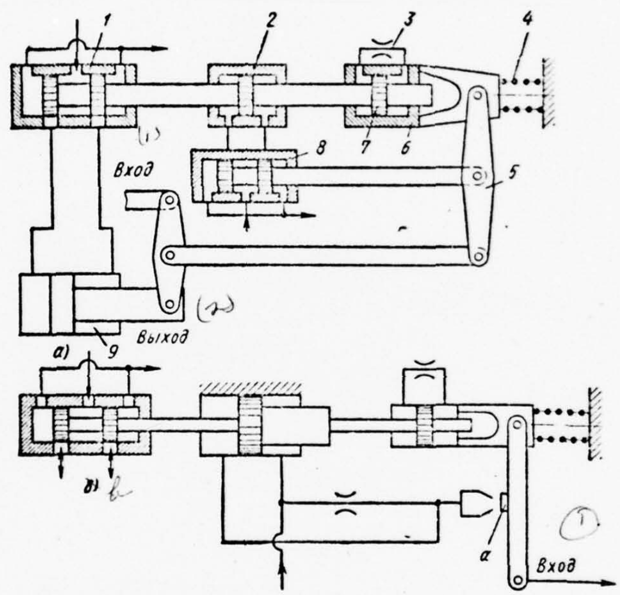
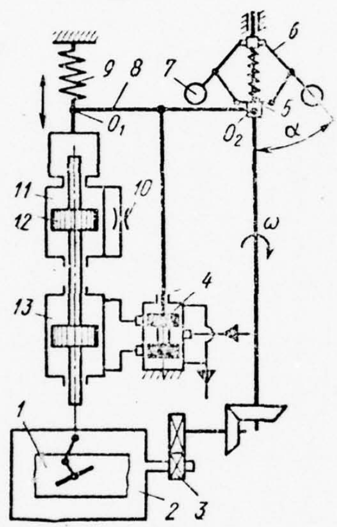


Fig. 146. Two-stage hydraulic booster with the correction of speed control of disagreement/mismatch.



Therefore is corrected the position of distribution valve 1 in accordance with input signal by the load of output/yield.

After a change in the disagreement/mismatch will cease itself, spring 4, straightening, will extrude/exclude liquid from the right cavity of cylinder 6 and will return it on the cessation of a change in the disagreement/mismatch at starting position. As a result auxiliary valve 8 and, consequently, also basic distribution valve 1 will engage the assigned positions, i.e., the position of equilibrium of system it is reduced.

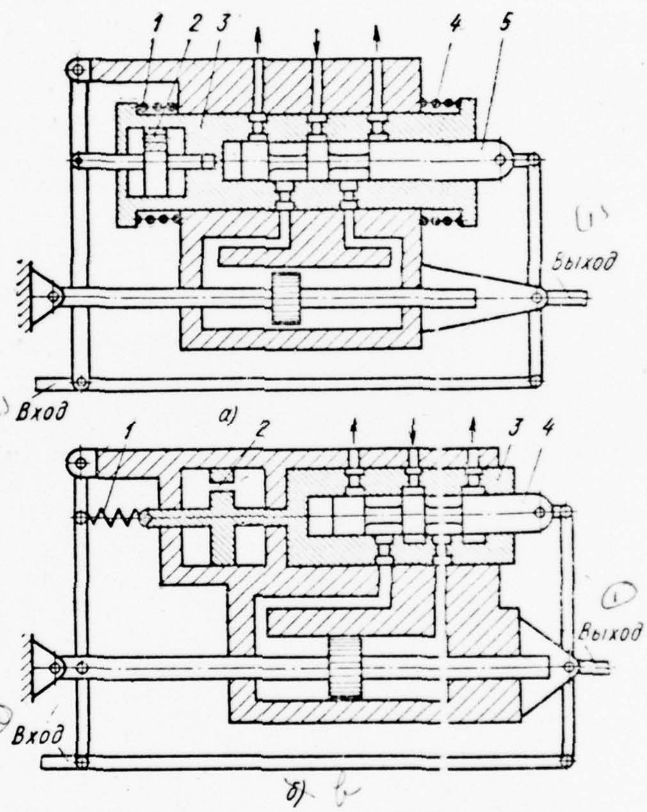
Thus, the distribution valve of this schematic can occupy certain determined position in accordance with the load of output/yield, not depended on the position of entrance.

In Fig. 145b is shown the circuit of two-stage amplifier, which differs from preceding/previous only in terms of the fact that the first cascade/stage of intensification is carried out in it in the form of nozzle - shutter/valve a.

Figure 146 depicts the schematic diagram of the hydraulic system of the automatic frequency control of the rotation of heat engine 2 with similar flexible feedback. During a change in the rate of the output shaft of engine in comparison with that which was assigned, the loads of 7 ball governor 6 change their position, characterized with angle α.

Fig. 147. Hydraulic boosters with derivative control.

Key: (1). Output/yield. (2). Entrance.



In accordance with this, is moved the clutch of 5 regulator, which causes rotation around point O_1 lever 8 and the displacement of connected with it valve 4 of performing actuating cylinder 13, which affects the controlled system (by changing the position of shutter/valve 1, which controls fuel consumption). As a result the controlled parameter (exit velocity) of engine 2 it is reduced.

The flexibility of described communication/connection is caused by the presence of spring 9, connected with the cylinder of 11 liquid dampers (cataract).

As a result of the resistance of throttle/choke 10 cylinder the cataract in the beginning of the piston stroke of 12 actuating mechanisms (hydraulic cylinder) is moved together with it, dilate/extending or compressing spring 9 and turning lever 8 around point O_2, i.e., to the moment the feedback in question returns gradually to the initial position, displacing the liquid through throttle/choke 10 of the one cavity of cylinder 11 into another.

The schematic of a similar mechanism with derivative control is shown in Fig. 147a. The bushing of 3 valves has the capability to be

moved in axial direction under the action of the force, transferred from damper 2, which reacts (because of throttle/choke) to the rate of change in the disagreement/mismatch. Springs 1 and 4 accept control signal from damper and on the cessation of a change in the disagreement/mismatch return bushing 3 to free position relative to the plunger of 5 valve.

Furthermore, in it are carried out two functional operations at the error signal: automatically are considered two functions ("proportionality" and "on derivative") is conducted the resulting effect.

Page 199.

Structurally this is reached by valve, discovery/opening flow areas occurs in moving the plunger of valve relative to housing (proportional control) and in moving the very bushing of valve (derivative control). These functions in valve are added either are deducted with respect that, increases or decreases disagreement/mismatch.

In such a case, when in hydraulic booster is not allow/assumed even insignificant constant disagreement/mismatch, necessary to ensure the action of adjuster also after disagreement/mismatch temporarily decreased to zero. The last/latter condition can be carried out by the introduction of effect on the integral of disagreement/mismatch with respect to time.

Figure 147b shows the schematic of the hydraulic booster in which the effect on integral, realize/accomplished by the introduction of spring 1 and of damper 2, is combined with proportional control. At first, when system is located in equilibrium and has zero disagreement/mismatch, bushing 3 and the plunger of 4 valves occupy the free position in which the valve is enclosed. When appears disagreement/mismatch, the plunger of 4 valves is moved relative to the housing of valve to the value, proportional to disagreement/mismatch, and causes the displacement of dash-pot piston 2 together with the bushing of valve. This displacement is proportional to the integral of disagreement/mismatch in time. As a result actual discovery/opening valve is determined by the mutual displacement of bushing and plunger of valve, which corresponds to the sum of the signals of proportional control and control on integral.

Every jet-edge unit (Fig. 148a) consists of a supply channel (nozzle) a and two outlet (combining) ducts (nozzles) c_1 and c_2 connected to a hydraulic engine (load). In addition, depending on the purpose of the equipment, it has a jet interaction zone (chamber) e and two control channels b_1 and b_2.

In combining nozzles the kinetic energy of fluid flow is converted into the potential energy (pressure), which develops the driving/moving effort/forces on the piston of the connected to this nozzle hydraulic engine. Depending on form, this jet-edge amplifier can execute both continuous and discrete operations. In the amplifier of proportional action (Fig. 148a) the jet, which escape/ensues of the channel of supply (nozzle) a, is divided in the absence of control signals equally between outlet ducts c_1 and by C_2, connected with load (hydraulic cylinder). During the supplying of control signal into one of the channels b_1 or b_2, the feeding jet will be deflected to the side, opposite to this channel, as a result the power at output/yield from the side of the deflected jet will exceed power at opposite output/yield. The characteristics of this amplifier depend on forms and processes of mixing in the chamber of the interacting jets and exchange of the amount of their motion.

The introduction of effect on integral raises the accuracy/precision cf work of hydraulic booster in transient conditions; however, it can lower its stability.

JET AMPLIFIERS

In automatic hydropneumatic equipment and in the systems of hydraulic steering find a use jet-edge amplifiers (devices) in which is utilized the coupling effect of the fluid flcws or gases between themselves and solids.

Jet-edge amplifiers are divided by two basic groups: continuous action (proportional amplifiers) and discrete action (double-stable amplifiers). Are carried out they both with the channels circular and rectangular cross sections.

Amplifiers of proportional action.

Fig. 148. Schematic diagram of the action of proportional type jet-edge amplifier.

Key: (1). Outlet pressure. (2). Right output/yield. (3). Left output/yield. (4). Differential pressure of control.

Fig. 149. The design diagram of proportional type jet-edge amplifier.

Key: (1). Receiving channel.

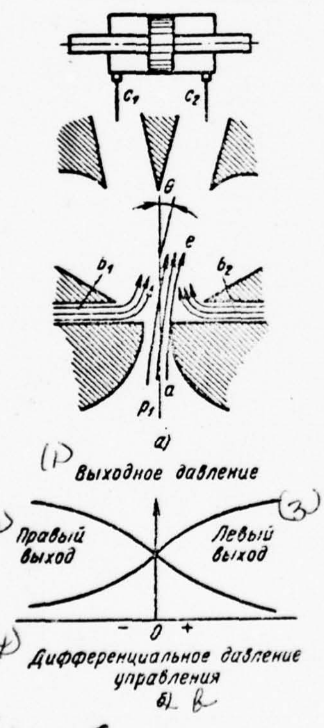


Fig. 148.

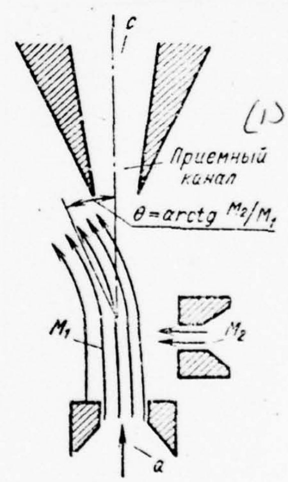


Fig. 149.

The output of a similar jet-edge amplifier is determined by difference in the expenditures, and also by the momenta of flow and by the pressures, which develop in its receiving (exit) channels c_1 and C_2 depending on load at output/yield. All these values depend on rate distribution of the deviated flow of the differential pressure of equation is given in Fig. 148b.

For desensitization of amplifier to changes of the resistance of load (pressure in the receiving channels c_1 and C_2 depending on load at output/yield. All these values depend on rate distribution of deviated flow and the angle θ to which it is deflect/diverted. Dependence of outlet pressure on the differential pressure of the administration is given in Fig. 148b.

For decrease of sensitivity of amplifier to changes in the load impedance (pressure in the receiving channels c_1 and C_2) chamber e with opening/apertures is connected with the atmosphere. When the receiving (load) channels c_1 and C_2 are completely overlapped (for example with the cessation of the piston of hydraulic cylinder), the flow through these of openings can escape/ensue into the atmosphere.

Jet deflection is realize/accomplished by different methods, and specifically, by the governing flow, applied through the governing channels b_1 and b_2, arrange/located perpendicular to the axle/axis of the feeding nozzle a, or by means of the rotation of this nozzle.

Let us designate by $M_1 = \rho u^2{}_1 \omega$ the momentum of the feeding liquid jet by section ω and by the density ρ, which escape/ensues of nozzle a at rate u_1. Let the governing jet with momentum M_2 be directed square with the feeding jet, the momentum M_1 the feeding jet considerably exceeding the momentum of the governing jet M_2 (Fig. 149).

According to the theorem about the conservation of momentum, the feeding jet under the action on it of governing jet (flow) will be deflected on angle

$$\theta = \mathrm{arctg}\,\frac{I_2}{I_1}.$$

With coaxial location feeding a and receiving c of channels and with zero control signal the feeding jet falls into receiving channel.

Page 201.

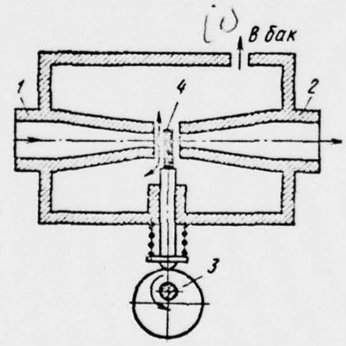


Fig. 150. Schematic of the jet-edge oscillator of pressure.

Key: (1). In tank.

During the supplying of governing jet, the feeding jet is deflect/diverted, and with in the governing pressure increase increasingly less part of the feeding flow falls into receiving channel. Experience/experiment and the calculation show that the insignificant deflection of the flow of control causes considerable changes in a difference in the expenditures through the outlet ducts of amplifier.

The form and the size/dimensions of chamber a of proportional amplifier in which interact manager and feeding the flows, must be carried out in such a way that the flow of supply would not undergo the effect of its walls.

Jet-edge amplifiers possess a series of qualities, which make it possible to apply them under the severe conditions of work, and in particular in wide temperature range, on the high level of vibrations and accelerations, and also with large impact loads.

Because of the absence in these amplifiers of their moving elements it is possible to manufacture from the materials, which ensure work in such specific conditions as nuclear radiations and

very high and low temperatures (fluid-jet elements are applied for work in the very broad band of temperatures from -196 to 980°C). During the application/use of approprate materials, they allow/assume work with aggressive liquids.

Work substances can be both liquids and gases, whereupon the devices, designed for work on gases, they can work also on liquids.

On operating speed jet-edge amplifiers exceed all the existing similar devices of mechanical type. The maximum frequency of the changeover of gas jet-edge amplifier cell/elements reaches several kHz. In work on low-viscosity liquids (water) the operating speed of fluid-jet elements as a result of a difference in the densities of liquids and gases approximately by an order is lower than operating speed in work in air; however, it is located at the level, which satisfies practice. Liquid generators work in the range of frequencies 150-5500 Hz.

The basic difference in the operating speed of hydraulic pneumatic jet-edge devices depends on the rate of the feeding jet and inertia properties of governing channel.

With the aid of jet-edge amplifier it is possible to obtain the pressure impulses of high frequency, including the periodic momentum/impulse/pulses, which reproduce varying loads. This makes it possible to utilize jet-edge technology for generation in the vibration testing units of high-frequency oscillations, and also for the imitation of intermittent loads on the power organ/controls of different machines during frequency and resource tests. It is represented also by the possible to change these loads according to predetermined program.

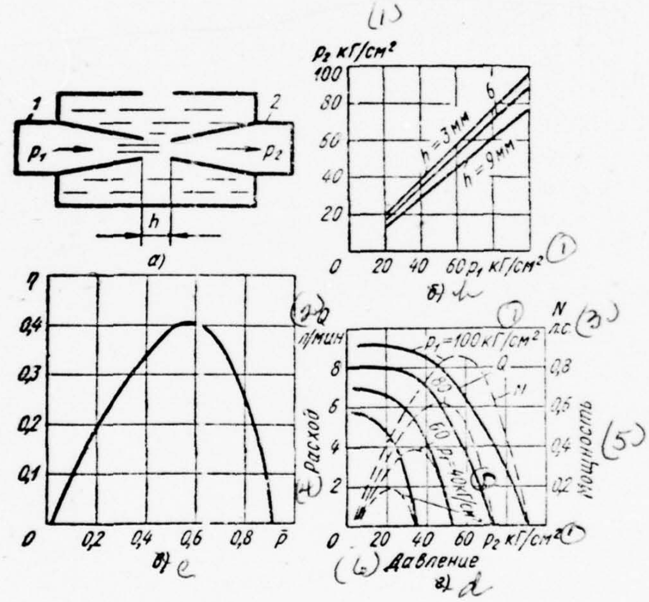
The schematic of the jet-edge oscillator of pressure is given in Fig. 150. Liquid under pressure will be fed to feeding convergent nozzle 1 and through combining nozzle 2 enters the actuating mechanism (hydraulic engine) of testing unit. The feeding channel of convergent nozzle can overlap with shutter/valve 4, led to oscillatory motions with the aid of shaped eccentric 3, the rotational speed of which determines the frequency of the overlaps by the shutter/valve of the feeding channel and the oscillation frequency of pressure in receiving sole 2. The law of a change in the pressure (shape of pulse) is assigned by the form of shutter/valve.

The amplitude of fluctuation of pressure in combining nozzle is assigned by pressure in the feeding nozzle and regulation of the cam.

[redacted].

Fig. 151. Schematic of the chamber of jet-edge proportional amplifier and characteristic of expenditure at output/yield.

Key: (1). kgf/cm². (2). 1/min. (3). hp. (4). Expenditure. (5). Power. (6). Pressure.



The tests showed the possibility of obtaining the stable amplitude of fluctuaticns of pressure to 200 kgf/cm² sinusoidal form at oscillation frequencies to 50 Hz.

The studies of the amplifier, made by the schematic, given in Fig. 151a, they showed that with the coaxial location of that feeding 1 and receiving 2 nozzles and their corresponding structural/design execution, and also the corresponding distance between their section/shear of the transferred energy loss relatively small. So, at applied pressure p_1 = 100 kgf/cm² and with distance h between nozzle edges 3-5 mm (diameter of the feeding nczzle 1 mm and receiving - 1.5 mm) lcss of pressure did not exceed with the zero expenditure of combining nozzle 3-4 kgf/cm².

Figure 151b gives the curves of the dependences of outlet pressure p_2 on input p_1 for the different distances h between nozzles. The minimum energy losses during flow of liquid in the conically tapering channel (convergent nozzle) cccur at angles of taper $alpha_1$ = 13-15°, also, in that which is diverging - with $alpha_2$ = 6-8°.

Figure 151d gives the curves (solid lines) of fluid flow rate at output/yield from combining nozzle 2 in pressure dependence p_2 loads (at the nozzle outlet 2) for the different applied pressures p_1 with the distance between nozzle edges h = 6 mm.

In accordance with change depending on the load (pressure p_2) of expenditure, changes also power N, reaching the maximum value with the determined load. The curves of the indicated dependence of power on pressure (load) p_2 for the conditions in question are given in Fig. 151d (broken lines).

From the aforesaid it follows that the nonlinearity of the expenditure in the function of load is analogous to the nonlinearity, which occurs with spill with an alternating/variable pressure differential of the throttle/chokes of constant section. With an increase in the distance between nozzles, the nonlinearity of expenditure is raised. It is obvious, if combining nozzle is connected with hydraulic engine (power cylinder), then the dependence of the exit velocity of the stock/rod of the latter will take the form of these curves.

Fig. 152. Schematic diagram of the action of discrete type jet-edge amplifier.

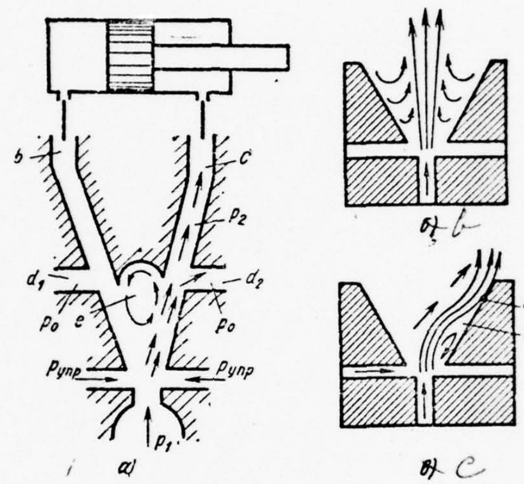


Figure 151c shows the curve of the full-load saturation curve of nozzles in a dimensionless form at different pressures in the feeding nozzle (from 40 to 100 kgf/cm²), the showing dependence the efficiency of fluid-jet element on the dimensionless parameter of load

$$\bar{p}=\frac{p_2}{p_1}.$$

Tests are carried out with the nozzles of the following size/dimensions: $d_{num} = 1,03$ mm and $d_{npuem} = 1,56$ mm; the distance between nozzles h = 6 mm.

Amplifiers of discrete action.

For the flow distribution of liquid, are applied also the jet-edge amplifiers of the discrete (intermittent) action, which are

actually certain structural/design modification of the examined amplifier of proportional action. In this amplifier of the stream, it has only two stable positions (Fig. 152). The basic design difference of the last/latter double-stable amplifier from the examined amplifier of proportional action is the approach of the lateral chamber walls e of the interaction of jets, thanks to which is ensured the possibility of the "adhesion" of jets to one of the walls of this chamber. An amplifier of this type is carried out by usually planar (with the rectangular ducts and the rectangular chamber of the interaction of jets).

The operating principle of discrete jet-edge amplifier is based on the property of the boundary layer of free flow "to adhere" to the adjacent surface (effect of Coanda), whereupon in this stuck position flow it remains after the break-down of the control signal, which caused this adhesion.

The mechanism of "adhesion" can be illustrated by the schematic diagram, shown in Fig. 152b and c. coaxial (central) the axle/axis of channel the position of the jet between the walls, forming the mixing chamber, shown in Fig. 152b, is unstable, and jet under the effect of any even sufficient small effects will be deflected into one of the

sides (Fig. 152c). In this case, the sectional area, past which it can pass the ejected flow from that side to which was deflected the flow, decreases, and with opposite it will increase. As a result between jet and chamber wall, the interactions will be created as a result of the ejecting action of jet separating circulation zone with reduced pressure, and also will arise the transverse gradient of pressures, directed to the same side, as the initial jet deflection, under action of which occurs further jet deflection.

If near the deflected jet on the one hand proves to be wall, flow from the side of this wall it is detained, which will lead to the emergence of a pressure difference from both sides of the jet: near wall is formed area of reduced pressures in the flow of the compensating liquid, as a result of which the jet will join ("will stick") the wall.

Page 204.

If jet passes near wall, then it seemingly had attract/tightened to it. In this case, is establish/installed the equilibrium between the force, which attempts to level off jet in the initial direction, and

the force, which appears in the separating zone b of reduced pressure, which bends the particle trajectory of the liquid in jet towards wall.

The change-over (changeover) of jet from one wall to another will occur, when the described equilibrium is disrupted, i.e., when pressure in the separating zone b so will increase that will occur the separation of flow from wall. For this, it is necessary to feed to this separating zone certain amount of liquid under pressure, which will be reached, if we increase expenditure from the right governing channel. The dividing eddy/vortex in this case increases, and point a of the adhesion of jet is displaced downstream. After this point will move toward the end of the wall, jet is detached away from the right wall, it will move to opposite wall and will enter the second outlet duct.

The change-over of jet is caused by the fact that through the right governing opening/aperture enters a sufficient jet in order to complete the loss/depreciation of the medium, exhausted from the starboard of strings, whereas from opposite side of working medium it is expend/consumed more than it enters through the governing opening/aperture. As a result pressure from left side is lowered the

jet "will stick" to this wall of channel.

In view of the directionality of jet, the overflows of signal jet of one inlet duct in another are small, if only outlet duct is not completely blocked. In order to remove this overflowing also in the case of the blocking (overlap) of the supplied outlet duct (with large load), are applied the lateral opening/apertures d_1 and d_2, connected with the atmosphere (p_0), through one of which the jet in this case is headed for the atmosphere, instead of overflowing into another inlet duct.

By the selection of the corresponding configuration of chamber e of the interaction of jets and outlet ducts b and c (Fig. 152a) it is possible to create this "locking eddy/vortex", which completely will remove the rotation of flow of one outlet duct in another under all conditions of their loading up to their complete overlap (flow will escape/ensue in this case into the atmosphere). To invert of flow and to deflect it to opposite outlet duct possible in this case only by the application/appendix of opposite on sign governing flow.

Experience/experiment shows that in flow direction, for example

into the load (receiving) channel c, pressure in the second load channel b are equal or even somewhat lower than the pressure p_0 the environment, so that opening or channel b will not affect the flow rate and pressure in channel c. Amplification factor in pressure of work on anechoic chamber (with the overlapped outlet duct) reached during testing this amplifier on water of value

$$\frac{p_2}{p_{ynp}} = 8 \div 10.$$

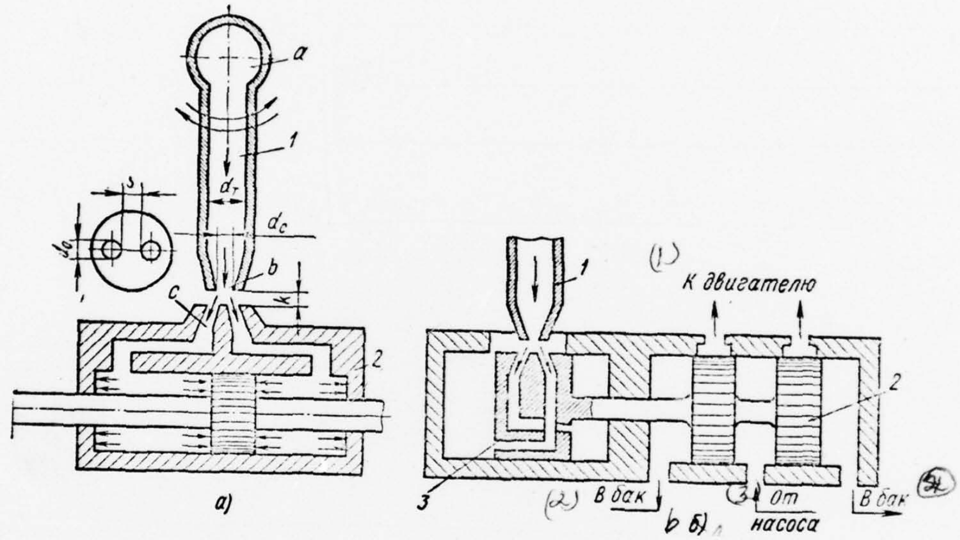
The flow rate of Q_{ynp} for control for the changeover of the discrete amplifier in question in work on water in practice does not depend on ratio p_1/p_2 and is equal $Q_{ynp} = (0,08 \div 0,09) Q_{num}$. Maximum relation of the flow rates of the output/yield of Q_{θ} and supply of the Q_n:

$$\frac{Q_a}{Q_n} \approx 0{,}9.$$

Time interval between the feed of control signal and the appearance of an output signal at the second output/yield of device (reaction time) depends on design features and the size/dimensions of this device, and also on the ratio of the pressure of the administration of p_{ynp} and supply p_1.

Fig. 153. Design concepts of jet-edge hydraulic boosters.

Key: (1). To engine. (2). In tank. (3). From pump.



For miniature gas devices this time does not exceed 40-50 microseconds, for water and working oils of hydraulic systems of hydraulic systems operating speed (with Re = 1000) 12-15 times lower than for gases.

Servo jet-edge amplifiers of proportional action.

The jet-edge amplifiers of proportional action are applied as distributors in servo type hydraulic boosters. Jet deflection realize/accomplishes in the latter case (Fig. 153a) by the rotation of the feeding tube (nozzle) 1, through which will be fed the liquid (feeding flow) to intakes c of actuating cylinder 2. The tube with 1 hinged end a is connected with the supply of power of system by the liquid which on leaving from nozzle b enters two arrange/located by a series intakes c of the distributive unit of actuating cylinder 2, each of which is connected with the appropriate cavities of the latter.

If the nozzle of jet pipe is arranged symmetrically relative to opening/apertures c, then pressures in both cavities of cylinder 2 are equal, and its piston is found in rest. During the bias of tube 1

relative to intakes the pressure into one cavity will exceed pressure in another, as a result piston will be displaced to the side. For an increase in the speed of flow for the target/purpose of an increase in the supply of kinetic energy, the tube usually is made in the form of conical nozzle (nozzle) with angle of 13-15°, the divergence angle of receiving channels 8-10°. By the systems of pneumatic automatics, the pressure usually does not exceed 8-10 kgf/cm²; however, in the systems of automatic hydraulic equipment, these devices in many instances work at pressures 200 kgf/cm² and above.

The feedback in diagram with the fixed piston (Fig. 153a) is ensured by the fact that cylinder 2 is moved to the side of the bias of tube 1 until is restore/reduced the disrupted symmetry of the position of nozzle relative to intakes c. The power stroke of performing hydraulic engine is determined into this case by the bias of tube.

The axle/axis of tube usually is arrange/located vertically, thanks to which the reaction of liquid jet acts against gravitational forces, decreasing the frictional forces in the supports of tube.

The advantage of jet-edge distributor is the fact that communication/connection between its component/links is realize/accomplished only by a liquid jet, thanks to which tube do not affect the unbalanced forces examined above, which act in valve. Furthermore, this distributor differs in terms of small inertia and small friction of moving elements, which increases sensitivity and operating speed servo systems. A distributor of this type allow/assumes to 100 changeovers per second.

Fig. 154. Two-stage jet-edge amplifier with electromagnetic control.

Fig. 155. Jet-edge amplifier with derivative control.

Key: (1). Control.

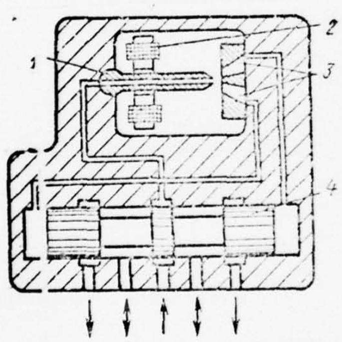


Fig. 154.

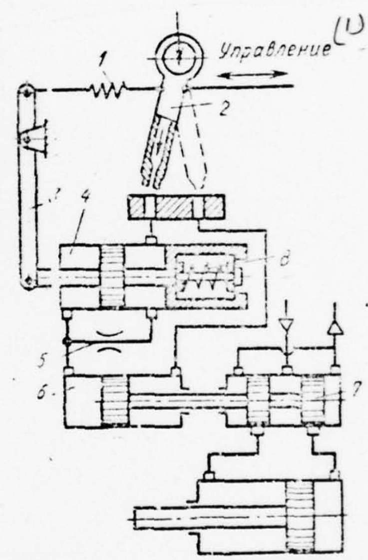


Fig. 155.

The dead zone of system with jet pipe usually is 0.01-0.02 mm. Interchangeable effort/force for the displacement of the tube of medium size is 3-5 g. Pressure in servomotor (in actuating cylinder) it is approximately 90o/o of inlet pressure (pressure, applied to tube).

An energy deficiency/lack in the jet-edge hydraulic booster is the fact that with low and zero control signals it has the high losses of the flow rate of supply. It is not difficult to see (see Fig. 153a) that in the free position of jet pipe (with zero signal) these losses of flow rate are equal to the maximum calculated flow rate of hydraulic engine. The maximum value the efficiency of jet-edge amplifier does not exceed aaaaaaaaaaaaa. In view of this the fluid-jet elements found use mainly as first stage of intensification in electrohydraulic amplifiers with feedback.

Two-stage jet-edge amplifiers.

Are applied also the two-stage diagrams, in which by the first cascade/stage of intensification serves jet pipe 1 and as the second, i.e., plunger 2, controlled by this tube (Fig. 153b). Intakes in this

diagram are made on piston 3, connected with plunger 2 of basic distribution valve, thanks to which the latter "follows" the tube which, in turn, "follows" the input signal.

Power loss to control comprises in one-stage system 15-18o/o of power, conducted to amplifier. In two-stage diagrams this loss, referred to the output power of hydraulic engine, composes the negligible fractions of this power.

In the systems of automation, the signal is form/shaped in the majority of cases with electromagnetic method. The circuit of standard jet-edge-slide-valve amplifier with electromagnetic control is represented in Fig. 154. The rotation of tube 1 is realize/accomplished with the aid of electromagnets 2. Combining nozzles 3 are connected with the right and left cavities of the actuating cylinder of basic valve 4, of the manager performing hydraulic engine.

Jet-edge amplifier with derivative control.

For an increase in the stability of servomechanism, are applied the circuits of servo slide valve with the jet-edge amplifier, controlled on derivative (Fig. 155). Liquid from tube (nozzle) 2 is headed not directly for the hydraulic cylinder of 6 drive of basic valve 7, but through cylinder 4 the so-called P I device (see also Fig. 145). In free position the piston of P I device is establish/installed by spring 8.

Page 207.

Let us assume that jet pipe 2 on with the action of steering signal input and spring 1 moved to the left and liquid began to enter the right cavity of the cylinder of 4 P I devices, displacing its piston and causing the admission of liquid into the cylinder of 6 basic valve. To the exchange of valve 7, jump/drop in the piston pressure of P I device, caused by compression of spring 8, causes the overflowing of oil through throttle/choke 5 of the right cavity into left. After this piston under spring effect 8 will return into free position, the liquid which is past through throttle/choke 5, will move the piston of valve and, consequently, also nozzle 2 at new position (shown broken lines). Lever 3 is supplementary feedback.

Selection of the operational characteristics of jet-edge amplifier.

The relation of diameter d_0 intakes in distributive unit (see Fig. 153a) to the diameter of the d_c of the outlet of the nozzle is usually equal of $d_0/d_c = 1,4$.

The distance between the section/shear (end/face) of nozzle and intakes in distributive unit for the hydraulic devices is usually equal of $4d_c$.

The diameter of intakes c in one-stage amplifiers is equal in the majority of constructions d_0 = 2-2.5 mm the diameter of the nozzle of $d_c \doteq 1,5 \div 2$ mm. The distance between intakes in the majority of constructions is 0.2-0.5 mm. With the indicated relationship/ratios of size/dimensions, the nozzle in the average/mean (neutral) position of tube does not overlap intakes. In some constructions is accepted the dependence

$$2d_0 + s \leqslant d_c.$$

The size/dimension of the outlet of nozzle is selected taking into account the provision for the required power of device, whereupon by two-stage devices the size/dimension of this opening/aperture usually does not exceed 0.5 mm.

Page 208.

Chapter V.

Circuits of Standard Hydraulc Systems their [redacted]/elements and basic calculations.

CIRCUITS OF STANDARD HYDRAULIC SYSTEMS.

The hydraulic system consists of energy source, such usually appears pump, actuating mechanism (actuating cylinder or hydraulic motor), and also the equipment for control of the fluid flow and protection of system from overloads. Specifically, necessary for the majority of hydraulic systems apparatus is the distributor of liquid, in function of which enters the provision for a direction of fluid flow to the working cavities of actuating mechanism. In complex systems can be applied several actuating mechanisms and energy sources, and also automatic control units, a control and an automatic effect on the law of the motion of the exit component/link of hydraulic engine.

The systems of any complexity are completed from elementary systems and their combinations. In view of the practical unlimitedness of the possible combinations of such elementary systems from which, are completed the more complex hydraulic systems of diverse machines and installations, let us be limited only to the description of the most standard network elements and their combinations which are applied practically in all machines.

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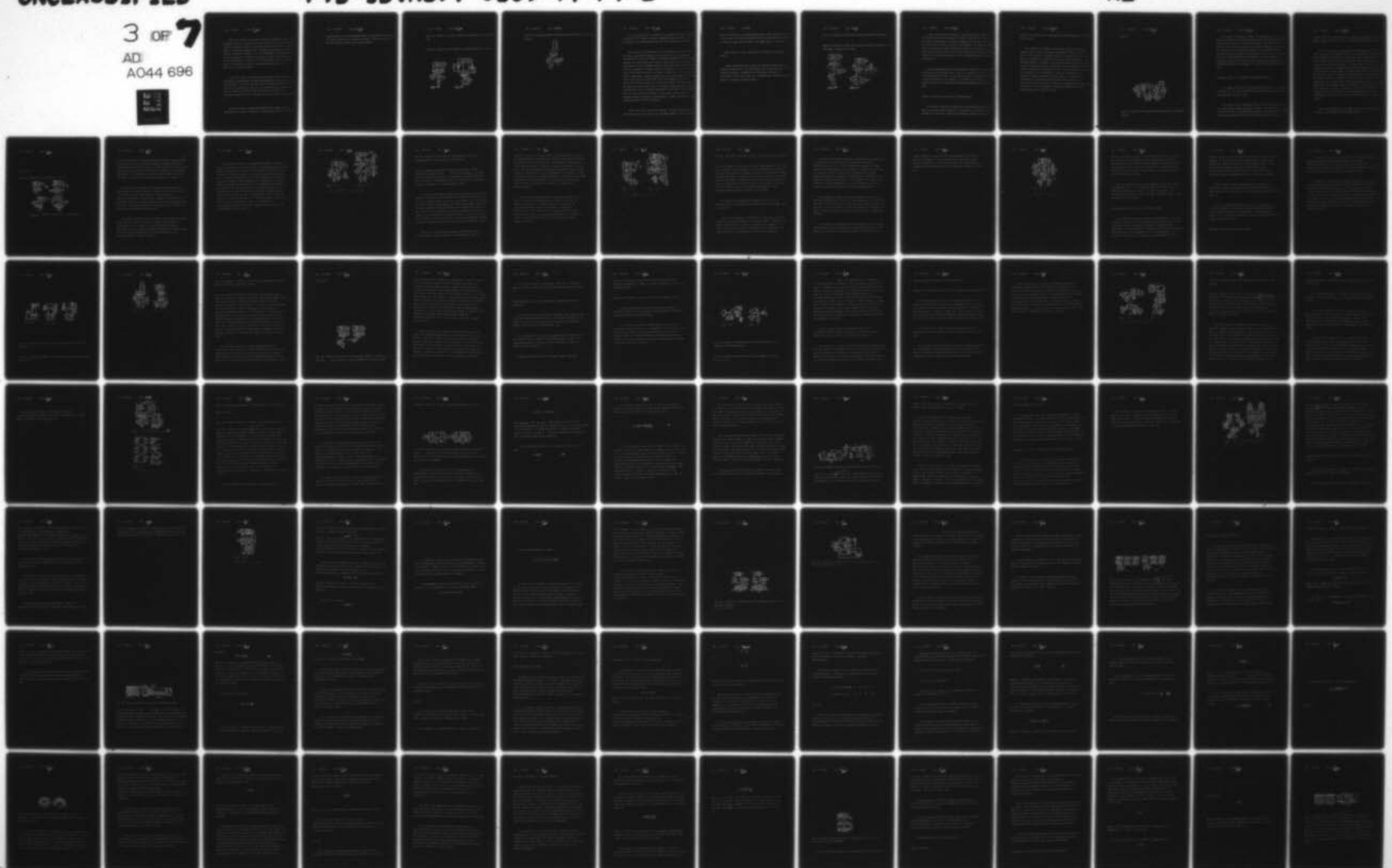

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Figure 156 depicts the standard circuit of hydraulic system with adjustable pump 3, rotated with electric motor M, with three-position fourway distributor (valve) 2 with the manual control with the aid of which is realize/accomplished the reverse of the piston of actuating cylinder (hydraulic engine) 1. In the mid-position of distributor 2, all its channels are connected with tank 5, which corresponds to the idling (discharging) of pump and to the "floating" state of piston. Pump 3 is supplied by filter 4, established/installed on inlet tubing, and by safety valve 6.

Figure 157 depicts the circuit of hydraulic system with the adjustable throttle/choke, established/installed in the line of feed (at entrance). In circuit provided for the connection/compound of the cavities of cylinder (see also Fig. 27a 28 and 45d), for providing what is applied the meltted down with the aid of detents 4 on stock/rod cylinder (or the connected with it any part) fourway switch 5.

System switches on uncontrolled pump 9 with safety valve 7, three-position fourway distributor 6 with manual control, adjustable

throttle/choke 2 and two-position switch 5 with drive from detent 4 of driving stock/rod of actuating cylinder 3 and with unit in the initial (upper) position under spring effect.

Fig. 156. Circuit of standard hydraulic system with the adjustable pump.

Fig. 157. Hydraulic system with the throttle governor of speed.

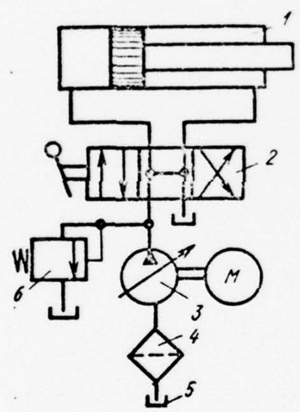


Fig. 156.

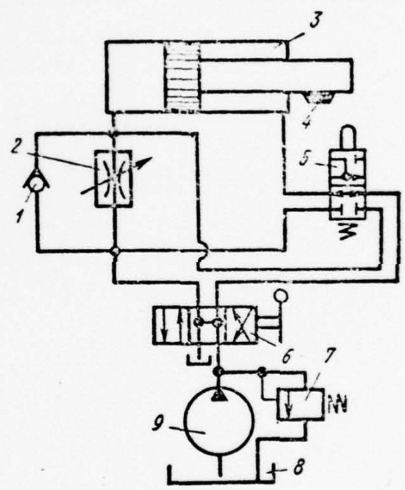
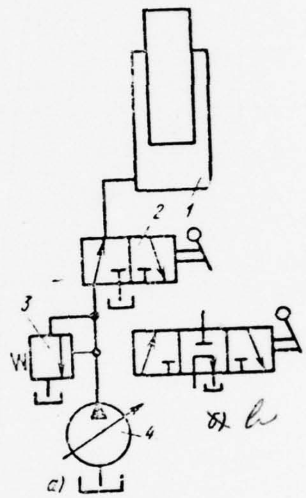


Fig. 157.

Fig. 158. Hydraulic system with the actuating cylinder of one-sided action.



In the mid-position of distributor 6, presented in Fig. 157, all its channels are connected between themselves and with tank, which corresponds the discharging of pump and the "floating" of piston.

The position of distributor in its left position (liquid enters the being intersected channels of the right field of distributor) corresponds to the piston stroke of actuating cylinder 3 to the right (liquid from pump enters left cavity), whereupon in this position of distributor b and of embedded switch 5 liquid both from the pump and from the nonoperative (right) cavity of cylinder 3 it enters its left cavity (the effective area of cylinder in this case is the sectional area of stock/rod), which corresponds to the quick traverse of piston to the right (see also Fig. 28a). After the pressure of detent 4 switch 5 will cease itself, it under spring effect will move upward and will cut off the left cavity of cylinder 3 from the right, after connecting the latter through distributor 5 with tank 8. As a result into the left cavity of cylinder 3, will enter only the liquid, passing through adjustable throttle/choke 2, which corresponds to the adjustable power stroke of piston 3.

During the setting up of distributor 6 at the right position, the liquid from pump 9 enters with unmounted switch 5 into the right

cavity of cylinder 3, realize/accomplishing a back stroke of piston. In this case, the liquid, displaced from the left cavity of cylinder 3, enters through throttle/choke 2 and check valve 1 into tank.

With pressure in this case switch 5, channel of pump will overlap.

Figure 158a depicts the circuit of hydraulic system with the actuating cylinder of 1 one-sided action and adjustable pump 4. The hydraulic system is controlled by three-stage two-position distributor 2 with hand drive. For a preservation from overloads, the system is supplied by safety valve 3.

Fig. 159. Hydraulic system with two-stage (pilot) distribution.

Fig. 160. Hydraulic system with two-stage (pilot) distribution and the manual discharging of pump.

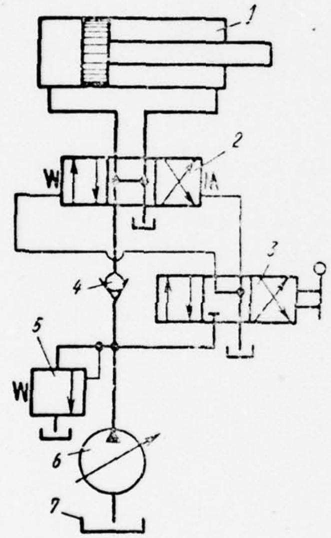


Fig. 159.

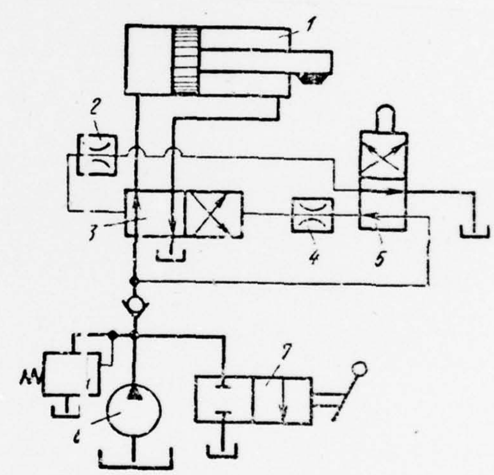


Fig. 160.

In the position of distributor 2, presented in Fig. 158a, the liquid from pump enters actuating cylinder 1. The line of tank in this case is overlapped. In moving distributor at opposite position, the outlet duct of pump 4 overlaps, and cylinder 1 is connected with tank, as a result the piston under the action of the weight of the given node is omitted down. The speed of depression is regulated with the aid of the throttling/choking of the abstract/removed liquid by distributor 2.

During application/use in the last/latter schematic of three-way three-position distributor (Fig. 158b) it is possible to ensure in its average/mean position the closing of liquid in actuating cylinder 1 (for the retention, for example, of the load in the built up position) during the simultaneous connection/compound of pump 4 with tank.

Hydraulic systems with two-stage intensification.

In automatic systems are common the two-stage distributors, in which the setting device affects the distributor not directly, but through the intermediate auxiliary distributor (pilot), thanks to

which it is possible it is substantial to lower the power of signal (see Fig. 50).

The schematic of hydraulic system with actuating cylinder 1, equipped with a similar two-stage distributor, which consists of basic 2 and auxiliary 3 fourway valves, is represented in Fig. 159. System is supplied by adjustable pump 6, and also protective 5 and reverse/inverse 4 by valves. Basic three-position fourway distributor 2 with the negative overlap of channels in the mid-position of this auxiliary distributor, presented in Fig. 159, the working cavities of the cylinders of the servodrive of the basic distributor are connected between themselves and with tank 7. As a result this distributor is establish/installed under spring effect in the mid-position by which all its channels are connected with tank, which corresponds to the discharging (translation/conversion into the mode/conditions of idling) of pump.

Page 211.

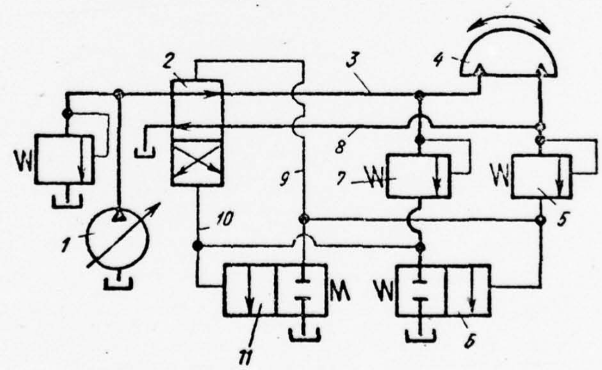


Fig. 161. Hydraulic system, which ensures rotary- oscillatory motions.

The diagram of analogous system is represented also in Fig. 160. System is supplied by uncontrolled pump 6 with the manual discharging (translation/conversion of pump into idling), realize/accomplished with the aid of two-stage two-pass distributor (switch) 7. Reversing the piston stroke of actuating cylinder 1 is realize/accomplished with the aid of the detents, established/installed on its stock/rod, acting on fourway two-position distributor 5, which ensures the changeover (reversing) of basic fourway two-position distributor 3. The switching rate of distributor 3 is limited by throttle/chokes 2 and 4.

Hydraulic systems of continuous (oscillatory) motion.

In a number of cases (in the machine tools, etc.) it is required to ensure the continuous (oscillatory) rectilinear or rotary motions of performing hydraulic engine.

The diagram of this hydraulic system with the hydraulic engine of rotary action (moment hydraulic cylinder) 4 is given in Fig. 161. Control of system is realize/accomplished by the automatically acting two-position distributor by 2 and discharging sequence valves 6 and

11 with control with the aid of the pressure of the liquid, passed by safety valves 5 and 7 at the end of each course of rotary piston.

In the position of the apparatuses of system, presented in Fig. 161, the liquid from adjustable pump 1 comes through the two-position hydro-controlled distributor 2 and pressure main 3 into cylinder 4 and is driven out from the latter into the tank through main line 8. At the end of each piston stroke of rotary cylinder 4, valve 7 as a result of a pressure increase will pass liquid into line 10 controls of distributor 2 and by valve 11, moving their working cell/elements. In this case, valve 11 connects the control line 9, connected with the upper cavity of distributor 2, with tank as a result of which distributor 2 is changed over, connecting pump with main line 8, that drives into the opposite cavity of cylinder 4. In this case, occurs the reverse of the latter, whereupon at the end of the course of cylinder enter into action the same sequence protective 5 and discharging 6 valves, which ensure the repetition of the reverse of piston.

The examined diagram is used also for the rotary oscillatory motions of the cylinder of rectilinear motion.

Pages 212-234.

Hydraulic systems with electromagnetic control.

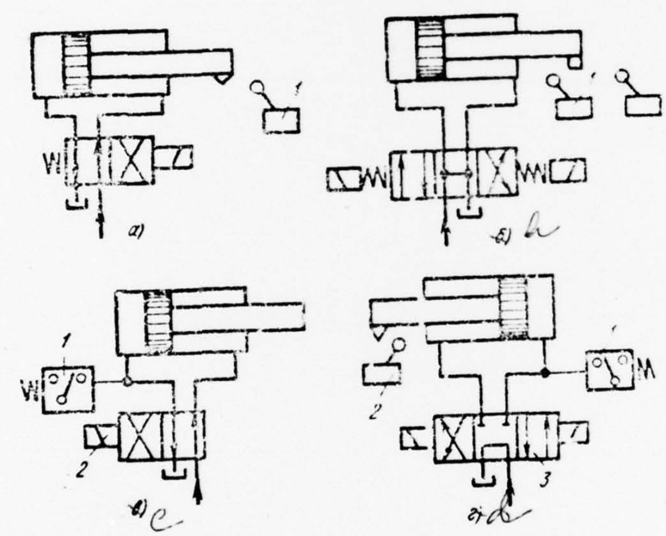


Fig. 162. Hydraulic systems with electrohydraulic control.

To the wide development of hydro-automation contributed application/use in distributive and other command hydraulic equipment/devices of electromagnetic (solenoid) drive. Are applied systems with one (Figs. 162a) and two (Fig. 162b) electromagnetic actuators of the movable cell/element of distributor. The return to the initial position of distributor in diagram in Fig. 162a is realized by a spring.

Governing electric signal in systems with distributor with electromagnetic actuator usually is supplied with different means to end electric switches (for example, to the limiters of displacement/movement) 1 (Fig. 162a), adjusted consecutively in the electrical circuit of launching/starting, and by the disconnection or the effect of operating pressure on the sensors in the diagram as on the subsequent diagrams, are not shown).

The circuit diagram of pressure relay 1 into hydraulic hydraulic system over the fixture of relay it breaks the feed circuit of electromagnet 2, as a result the spring translate/transfers distributor into the initial position. With the aid of springs occurs the setting up of distributor to the mid-position and in the diagrams, presented in Fig. 162b and d.

Limit switch frequently is combined with pressure relay (Fig. 162d). The reverse of the piston stroke of actuating cylinder in this system is realized by end switch 2, which affects the latent of the stock/rod of actuating cylinder. The pump is disconnect/turned off (it is translate/transferred into the mode/conditions of idling) by the pressure relay 1, which disrupts at the termination of piston stroke the electrical feed circuit of the magnets of distributor 3, as a result it by internal springs is establish/installed in the mid-position, in which the cavities of cylinder overlap (they are blocked), and the line of pump is connected with tank. Figure 163 depicts the schematic diagram of the hydraulic system, in which the disconnection (discharging) of pump 4 is realized by an electromagnetic two-pass distributor (bypass) 3 at the signals of electrohydraulic relay. Page 213.

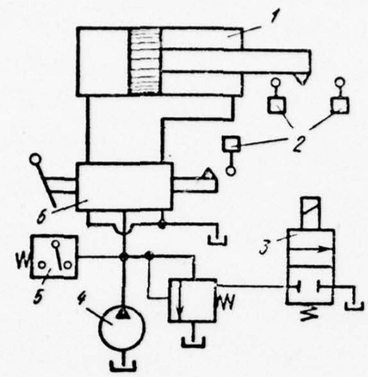


Fig. 163.

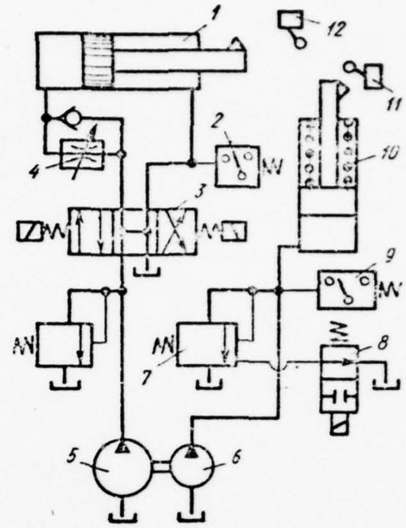


Fig. 164.

See following page

Fig. 163. Hydraulic system with the disconnection of pump by electrohydraulic pressure relay and end switches.

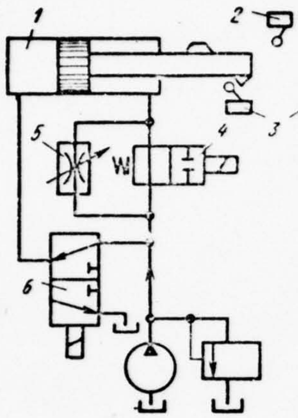
Fig. 164. Hydraulic system with control with the aid of limit switches and pressure relay. End electric switches 2 are given here into action by detents on the stock/rod of actuating cylinder 1 and the plunger of distribution valve 6 or with the aid of pressure relay t, connected to the delivery line of pump.

The diagram of hydraulic system with two pumps and the actuating cylinders, controlled by limit switches (limiters on position) 12 and 11 and to pressure relays 2 and 9, is represented in Fig. 164. System is equipped by two paired pumps, one of which (6) supplies cylinder 10, which fulfills auxiliary functions (for example, the cylinder of the mechanism of the terminal of article in machine tool), and the second (5) it supplies basic working cylinder 1. In the mid-position of distributor 3 and upper positions of two-pass distributor (bypass) 8 with electromagnetic actuator both pumps are connected with tank.

During the changeover of distributor 3 into one of the end positions pump 6 supplies liquid to the appropriate cavity of

cylinder 10. At the end of the piston stroke of cylinder 10 detent its stock/rod affects disconnection switch 11, with the aid of which is included the feed circuit of the electromagnet of distributor 3, of the manager cylinder 1. Since the pressure in cylinder 10 at the end of the course grow/rises, wear/operates pressure relay 9, which breaks the feed circuit of the electromagnet of distributor 3, which under spring effect is establish/installed in the mid-position, which corresponds to discharging pump 5. The velocity of the foreward stroke (course to the right) of cylinder 1 is regulated by throttle/choke 4.

one of the widespread systems of system with control of end electric switches and electromagnets is given in Fig. 165. The reverse of the piston of actuating cylinder 1, and also the connection/inclusion and the disconnection of adjustable throttle/choke 5 are realized here by two two-position two-pass distributors by 4 and 6, controlled end switches 2 and 3 from the detents, established/installed on the stock/rod of actuating cylinder. Page 214.



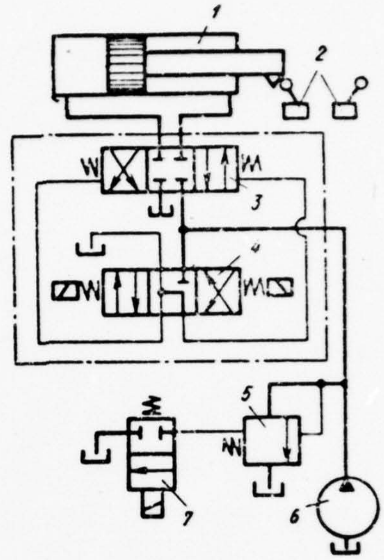


Fig 165 Fig 166

See following page.

Fig. 165. Hydraulic system with electric control from limit switches.

Fig. 166. Hydraulic system with two-stage electromagnetic control and limit switches. In the position of these distributors, presented in Fig. 165, both cavities of actuating cylinder 1 are connected between themselves through the channels of distributors 4 and 6, as a result into the left cavity of cylinder enters both liquid from pump and the liquid, displaced from the right cavity of this cylinder (see also Fig. 45d). This position of distributors corresponds to the accelerated piston stroke of cylinder to the right.

The speed of the displacement/movement of piston and the developed with it effort/force are determined by the sectional area of stock/rod.

During the changeover of distributor 6 to the upper position, realized by one of end switches 3, the left cavity of cylinder 1 is connected with tank, as a result the piston will be mixed to the left. The piston speed and the developed with it effort/force are determined by the sectional area of cylinder from stock/rod.

In certain intermediate position of stock/rod its detent acts on end switch 2, the clutch magnet of distributor 4, which is establish/installed in the right position and overlaps the bypass channel of throttle/choke 5, as a result the liquid will be be able to driven out from the right cavity of cylinder only through throttle/choke 5. This position of distributor 4 corresponds to the adjustable working piston stroke of the cylinder, with which it will develop the same effort/force, as in the preceding case, but the speed will be determined by the control of throttle/choke 5.

Electromagnetic control has special advantages when using the two-stage distributors, in which the electromagnet sets in motion the distributor of first stage of intensification (pilot). In view of the fact that in this case can be applied the miniature distributor, the power of electric signal for a control can be lowered to negligibly low value.

The diagram of the hydraulic system, controlled by end electric switches and two-stage distributor, is represented in Fig. 166. The steering of the piston of actuating cylinder 1 is realized by end

electric switches 2, which feed the electromagnets of auxiliary (pilot) distributor 4, of the manager basic distributor 3 with hydraulic changeover. To the extreme positions of distributor 4 corresponds power supply of one of the cavities of cylinder 1. Page 215.

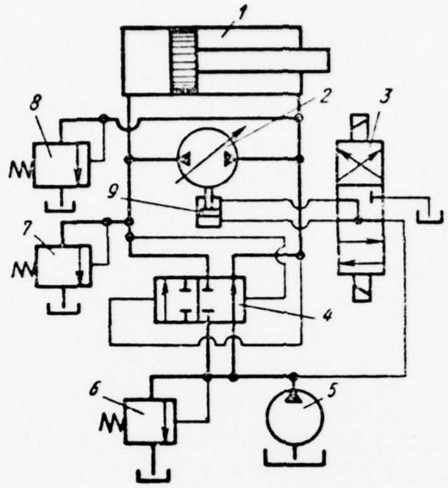


Fig. 167.
See following page.

Fig. 167. Hydraulic system with the adjustable reversive pump. During the setting up of distributor 4 to the mid-position, realized by springs during de-energizing its electromagnets, in the mid-position is establish/installed also basic distributor 3, blocking (cutting off) in this case of liquid in both cavities of actuating cylinder 1 and record/fixing its piston.

Upon the switching on of the electromagnet of two-pass switch (distributor) 7 line of the hydraulic steering of safety valve 5 is connected with tank, as a result this valve is converted into overflow, connecting pump 6 with tank (pump comes into action in the mode/conditions of idling).

Electrohydraulic systems with the adjustable pump.

To hydraulic systems with two-stage electrohydraulic control is related the system with the adjustable reversive pump whose reverse is realized by the servodrive, controlled by electrohydraulic distributor. A similar diagram of hydraulic system with the reversive adjustable pump 2 and hydraulic steering of productivity on the position of the piston of 9 servodrive is represented in Fig. 167.

System is equipped by booster pump 5, which feeds the control system (control) of the feed of the basic working pump, and which also realizes his makeup. Signal for the feed reverse of pump 2 enters from auxiliary fourway three-position distributor 3 with electromagnetic control, which obtains electric signal from the appropriate command equipment/device of the system of automation.

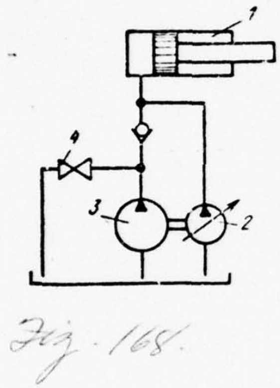
During reversing pump 2 simultaneously is changed over two-position fourway distributor 4 with hydraulic steering to the power supply by the booster pump of 5 corresponding suction cavity of pump 2.

Pump 2 is equipped by safety valves 7 and 8, adjusted to the required pressures with the direct/straight and reverse/inverse piston strokes of cylinder 1, while pump 5 - by safety valve 6, adjusted to the pressure, necessary for meeting requirements for the control system and makeup.

Hydraulic systems with two paired pumps.

In a number of machines, in particular in the machine tools, are common the diagrams with two paired pumps, one of which (uncontrolled) works on the low pressure, which requires for providing the accelerated idling, and the second (adjusted) works on the high pressure, which corresponds to working strokes.

The simplified diagram of this system is given in Fig. 168. The quick traverse of piston is provided by the total feed of two pumps of high (2) and low (3) pressures. At the termination of quick traverse pump 3 by hand or automatically over the signal of pressure is disconnected (in the diagram in question this disconnection/cutoff is realized with the aid of discovery/opening pet cock 4 by hand), whereupon the power supply of cylinder 1 is provided by one pump 2, which in the majority of diagrams is fulfilled that which is adjusted. Page 216.



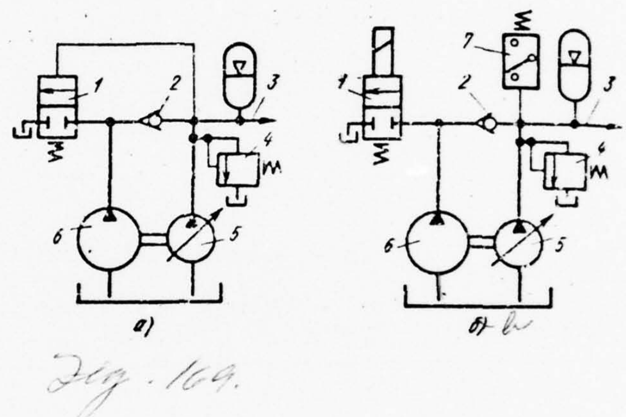


Fig. 168. Diagram of system with two paired pumps of high and low pressure.

Fig. 169. Diagrams of hydraulic systems with two paired pumps and gas storage tank.

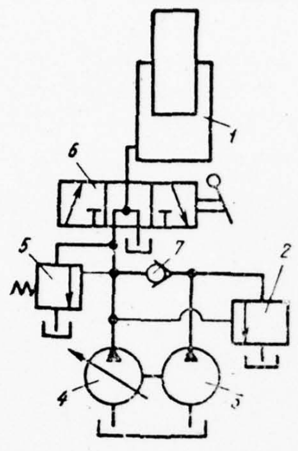


Fig. 170.

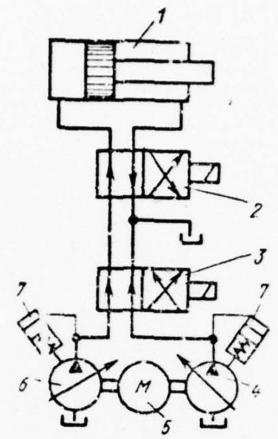


Fig. 171.

Fig. 170. Diagram of hydraulic system with two paired pumps and the actuating cylinder of one-sided action.

Fig. 171. Diagram of hydraulic system with two adjustable pumps.

Figure 169a shows the schematic diagram of a similar hydraulic system of the power supply of users by two paired pumps 5 and 6 with automatic changeover. Until pressure in the line of 3 users achieves the assigned value, at which is adjusted the return spring of two-pass distributor (switch) 1, the channels of the latter will be overlapped, and into line 3 enters liquid from both pumps. At the assigned pressure, determined by the characteristic of the spring of switch 1, the low-pressure pump 6 automatically will be switched to tank, pump 5 with the aid of check valve 2 it will disconnect from switch 1 and will continue the power supply of hydraulic system. The pressure, developed in this case with pump 5, is limited by safety valve 4.

The schematic diagram of a similar hydraulic system is represented in Fig. 169b. This diagram differs from in terms of the examined above fact that the disconnection/cutoff of the pump of 6 low pressure is realized by the electrohydraulic pressure relay 7, which feeds at the assigned pressure signal on electromagnetic switch

1. Page 217.

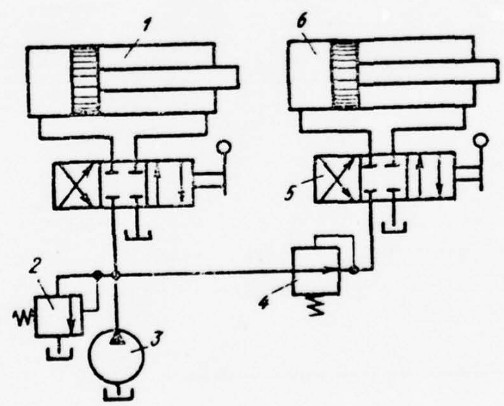


Fig. 172. Hydraulic systems with two actuating cylinders, supplied by one pump. The diagram of a similar system with two paired pumps

and one-sided actuating cylinder is represented in Fig. 170. Pump 4 of this system is designed to work with high, while pump 3 is designed at low pressure. System is equipped by three-position three-way distributor 6 with manual control and two valves, one of which (5) is the safety valve of high-pressure 4 pump, and the second (2), controlled by the pressure of pump 4, by the relief valve, which changes over with that which was assigned pressures to the assigned value cylinder is supplied by both pumps. Upon reaching of the pressure of the control of relief valve 2 pump 3 automatically is changed over to tank and the further power supply of cylinder 1 it is realized only by adjustable high-pressure 4 pump. For a disconnection/cutoff in this case of pump 4 from tank is applied check valve 7.

Figure 171 depicts the diagram of hydraulic system with two pumps of the alternating/variable feed with electric drive 5, one of which (4) low (large feed), and the second (6) high (fine feed) pressure. Pumps are equipped by the mechanisms of 7 control of feed on pressure. In the position, presented in Fig. by 171, pump 4 is switched to tank, and pump 6 supplies the left cavity of actuating cylinder 1. During the changeover of electromagnetic distributor 2 to the second position occurs the reverse of actuating cylinder 1.

In the left position of distributor 3 with tank is connected pump 6, and pump 4 supplies through distributor 2 actuating cylinder 1.

Power supply by one pump of two hydraulic engines with different pressures.

Figure 172 depicts the diagram of hydraulic system with one pump 3 and two actuating cylinders 1 and 6, one of which (cylinder 6) is designed to work with the external load (pressure), considerably smaller than the load of second cylinder (1).

For a decompression in the power-supply system of cylinder 6 to the required value, is applied the reduction valve (see Fig. 72), established/installed on the inlet into distributor 5. Pump 3 is equipped by safety valve 2.

Hydraulic systems with rotor hydraulic engines (hydraulic

motors) do not differ in principle from hydraulic systems with actuating cylinders. The control of speed is realized by a choke or volumetric method.

Systems of the throttle control of the speed of hydraulic motor.

The given information on throttle control with actuating cylinder is valid also for hydraulic systems (hydraulic drive) with the hydraulic motors of rotary motion.

In the latter case only one should consider some specific peculiarities associated with the possibility of the emergence of the large forces, caused by the high accelerations of the shaft of hydraulic motor in transient conditions. Accelerations usually are checked by the special automatic adjustable braking valves, adjusted on the gutter of motor. Page 218.

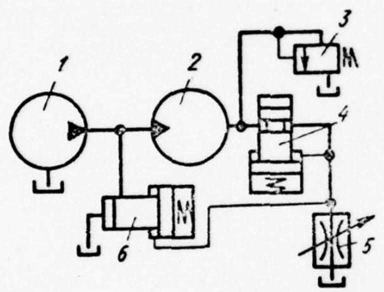


Fig. 173.

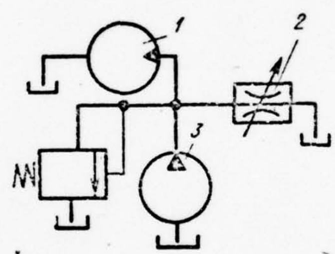


Fig. 174.

Fig. 173. Diagram of hydraulic system with the hydraulic motor, adjusted by differential valve.

Fig. 174. Diagram of hydraulic system with hydraulic motor and throttle control.

Figure 173 depicts one of the schematics of the throttle control of hydraulic motor 2. System is supplied by the pump of 1 constant feed with feed control by overflow differential valve 6, controlled depending on the drainage pressure of motor (see also Fig. 114). At output/yield from hydraulic motor 2 are established/installed braking valve 4 and adjustable throttle/choke 5. During the increase in the resistance of throttle/choke 5, caused by an increase in the velocity of motor, changes (it increases) the resistance of braking valve 4 (inner valve is omitted down) and simultaneously automatically changes (it decreases) the pressure of the adjustment of differential safety valve 6, established/installed on output/yield from pump.

Besides braking valve at output/yield from motor is established/installed also safety valve 3, adjusted to pressure higher than that, to which is adjusted braking valve 4.

The diagram of the simpler system of throttle control (with pump 3 of constant feed) is represented in Fig. 174. The exit velocity of hydraulic motor 1 here changes with adjustable throttle/choke 2, established/installed in parallel with hydraulic motor. With the aid of this throttle/choke part of the feed of pump is discaried (it is

abstract/removed) into tank (see also Fig. 102d).

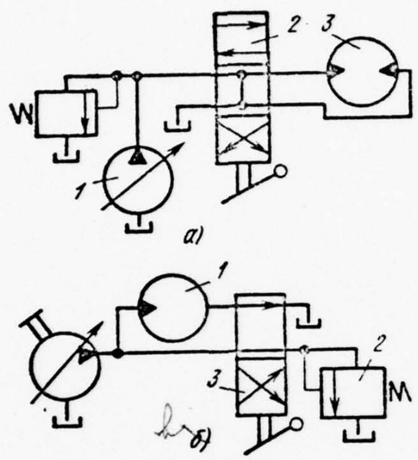
Systems of the volumetric control of the velocity of hydraulic motor.

In connection with the fact that the throttle control of the drives of rotary motion in question is accompanied as the control of the drives of rectilinear motion, by loss of power, is obtained the application/use of a system is common the control system of the exit velocity of drive with the use displacment pump and hydraulic motor, one of which or both are machines with the adjustable working volume.

The reverse of the output shaft of hydraulic motor can be realized with the aid of distributor or the reverse of the feed of pump.

The hydraulic system, in which the reverse of hydraulic motor 3 is realized with the aid of distributor 2, is represented in Fig. 175a. In diagram is applied adjustable pump 1. In the mid-position of distributor 2 all its channels are connected with tank.

Figure 175b depicts the diagram of hydraulic system with irreversive hydraulic motor 1, controlled by distributor 3, in function of which enters only the start of this hydraulic motor. In the drain line of hydraulic motor 1 is established/installed braking safety valve 2, which automatically is included during the translation/conversion of distributor 3 into the position, opposite presented in Fig. 175b (with the disconnection of the power supply of hydraulic motor 1). In this position the pump is connected with tank, and the motion of hydraulic motor brakes. Page 219.



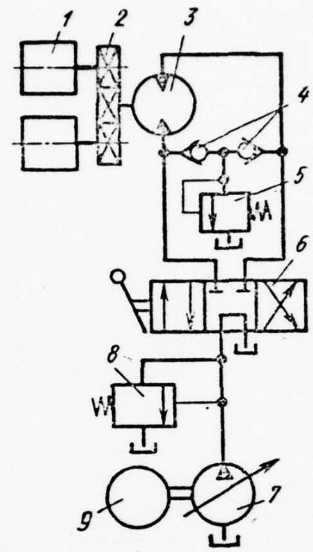


Fig. 175.
[See following pages]

Fig. 176

Fig. 175. Hydraulic systems with hydraulic motors and slide-valve control.

Fig. 176. Hydraulic system with the reversive hydraulic motor, controlled by three-position fourway distributor. [End of caption] The calculation of the pressure of braking valve conducts from the condition of the absorption of the kinetic energy of the rotating parts of hydraulic motor and connected to it mass of load. In the position of distributor 3, presented in Fig. 175b, the drain line of hydraulic motor is connected with tank.

Figure 176 depicts the diagram of hydraulic system with the reversive hydraulic motor, controlled with the aid of distributor 6. System is equipped by support valve 5, which acts in the work of hydraulic motor in both directions. For this providing in system are two check valves 4. This system is applied for the drive of the rolls of 1 rolling mill. The drive of rolls 1 is realized by reversive piston uncontrolled hydraulic motor 3 through pinion drive 2. Power supply is realized by the piston pump of 7 adjustable feed, given in rotation/revolution electric motor 9. The reverse of hydraulic motor 3 is realized by fourway three-position distributor 6 with manual control. Pump 7 is equipped by safety valve 8. Frequency of the

rotation/revolution of hydraulic motor in similar diagrams usually 5-100 r/min.

In the hydraulic systems of automatic control with reversive hydraulic motor widely are applied the diagrams with electromagnetic control.

one of such diagrams, in which is provided also braking hydraulic motor in both directions, is represented in Fig. 177. For this are applied differential valves 10 and 1, which are the combination of relief valve with the valve of counterpressure (braking), controlled by pressure $p_{вх}$ at the inlet into hydraulic motor.

The reverse of hydraulic motor 8 is realized by fourway three-position valve 6 with electromagnetic control and return springs. Upon the start of distributor to the power supply of main line 3 liquid through with pressure $p_{вх}$ inlet enters through governing main line 11 to controlled relief valve 10, holding its gate in the open position, in which the liquid from hydraulic motor 8 enters without resistance through main line 9 into tank.

With the disconnection of electromagnets valve 6 is establish/installed by return springs in the mid-position, in which pump 4 is connected with tank. Page 220.

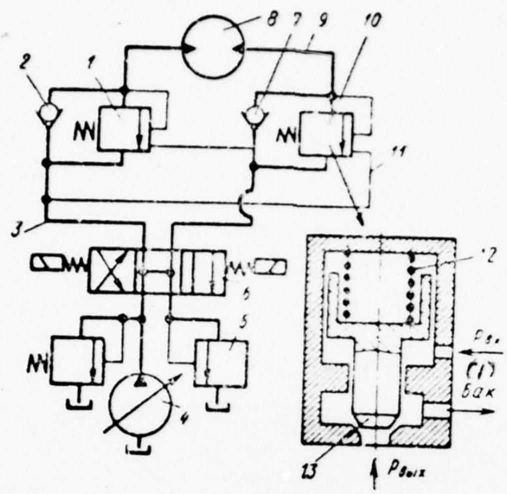


Fig. 177. [See following page]

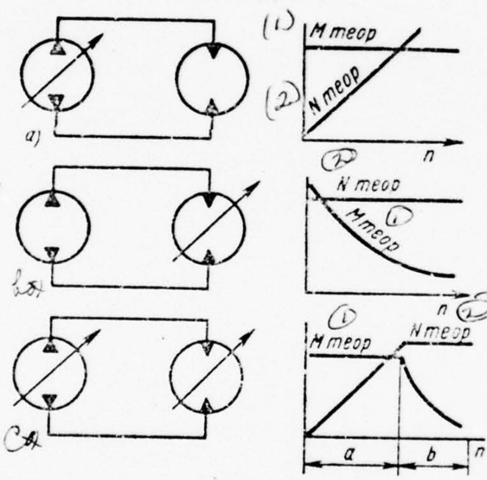


Fig. 178.
[See following page]

Fig. 177. System with reversive hydraulic motor and braking valves.

Key: (1). Tank.

Fig. 178. Control circuits of hydraulic drive of rotary motion.

End of caption.

Key: (1). M theory. (2). N theory. In this case the pump works only under the small (3.5 kgf/cm^2) pressure, created by support valve 5, necessary for the elimination of the air suction and cavitation. Since this pressure insufficiently for the retention of the gate of 13 relief valves 10 in the open position, this valve acts as pressure (braking) at output/yield from hydraulic motor 8, which in this case as a result of the inertia of its rotating parts works as pump. The output resistance from hydraulic motor is determined by the control of the spring of 12 this valves. The controlled/inspected braking in the work of hydraulic motor 8 in the opposite direction is realized with the aid of analogous valve 1 whose adjustment conducts independent of valve 10. Check valves 2 and 7 serve for the direct/straight circulation of flow into the circuit/bypass of valves 1 and 10.

In machine-building the propagation obtained mainly the

hydraulic systems, the velocity and the reverse of hydraulic motors in which it is realized by a feed control of pump and by a change in its direction (Fig. 178a). In these systems are applied both the uncontrolled (Fig. 178a) and adjustable (Fig. 178b) hydraulic motors. Are applied also diagrams with those which are adjusted by pump and hydraulic motor (Fig. 178c), during application/use of which is expanded the range of the control of velocity, and also the possibility of the selection of output characteristics of drive at the torsional moment M and according to power N (Fig. 178).

The diagram of system with adjustable reversive pump 2, the powered by electric motor by 1, and reversive uncontrolled hydraulic motor 5 is represented in Fig. 179a. System is equipped by safety valve by 8 and check valves 9 and 3, which ensure the connection/compound of valve 8 with the reverses of pump with delivery line. For the makeup suction barrels from tank 7 are applied check valves 4 and 6, being open/disclosed under action pressure differences and vacuum in the appropriate feed lines of pump.

More advanceed is the system with the enclosed circulation, in which the makeup of the suction line of basic (worker) pump 3 conducts by auxiliary (feed) pump 1 (Fig. 179b), overflow valve of 2

of which is adjusted to small (3-5 kgf/cm^2) pressure. Page 221.

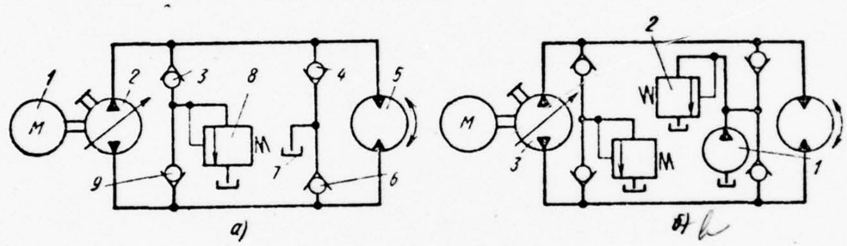


Fig. 179. Diagrams with the adjustable pumps and reversive hydraulic motors. Because of the application/use of a makeup of the suction line of pump under pressure is provided reliable filling with the liquid of its chambers.

The theoretical frequency of the rotation/revolution of hydraulic motor calculate for this case from the condition of the equality of the volumes, described by the working cell/elements of pump and motor (by pistons, by teeth, by blade/vanes, etc.) per unit time

$$Q_{T.\text{н}} = Q_{T.\text{м}} \quad \text{or} \quad q_\text{н} n_\text{н} = q_\text{м} n_\text{м},$$

where $Q_{T.\text{н}} = q_\text{н} n_\text{н}$ and $Q_{T.\text{м}} = q_\text{м} n_\text{м}$ - the volumes, described by the working cell/elements of pump and hydraulic motor per minute; $n_\text{н}$ and $n_\text{м}$ - the frequency of the rotation/revolution of pump and hydraulic motor under r/min; $q_\text{н}$ and $q_\text{м}$ - the pump displacements and hydraulic motor (calculated feed in one revolution).

Hence theoretical frequency of the rotation/revolution of the motor

$$n_\text{м} = n_\text{н} \frac{q_\text{н}}{q_\text{м}}. \qquad (63)$$

On the basis of expressions (63) and (61) it is possible to write for those which were distributed in the hydraulic systems the piston pump and the motor of the axial-piston types

$$n_{м}=n_{н}\frac{q_{н}}{q_{м}}=n_{н}\frac{\operatorname{tg}\nu_{н}}{\operatorname{tg}\nu_{м}}\frac{D_{н}d_{н}^{2}z_{н}}{D_{м}d_{м}^{2}z_{м}}. \qquad (64)$$

From the last/latter expression it follows that to regulate the velocity of exit drive shaft (motor) is possible in such a case, when one of the aggregate/units (pump or motor) adjusted, and, if adjusted is the pump, then theoretically it is possible to obtain the frequency of the rotation/revolution of motor from zero (it corresponds $q_{н}=0$) to the maximum (it corresponds $q_{н}=\max$) value. With both adjustable machines or at least with one adjustable motor theoretically it is possible to obtain the frequency of the rotation/revolution of motor from zero (it corresponds $q_{н}=0$) to infinite number (it corresponds $q_{м}\approx 0$).

The control of the exit velocity of piston type drive realizes by a change in the eccentricity ii of the angle of the slope of disk (see course "displacment pumps and the hydraulic engines"), which of pump can decrease to zero, and of hydraulic motor to certain minimum value, after which the mechanical efficiency of hydraulic motor sharply is reduced, as a result hydraulic motor it is converted into the self-braking system.

With the control of the exit velocity of the shaft of motor by a change in the working volume $q_н$ pump with constant working volume $q_м$ motor (see Fig. 178a) we will obtain (see expressions (36)] with a constant pressure differential of liquid alternating/variable power $N_{теор}$ and the constant torsional moment $M_г$ on the shaft of hydraulic motor (losses we disregard), while with the regulating of the working volume of motor with constant pump displacement (see Fig. 178b) - constant power $N_г$ and the alternating/variable torsional moment $M_г$.

In machine-building predominantly are applied the drives with the adjustable pump and the uncontrolled hydraulic motor. Page 222.

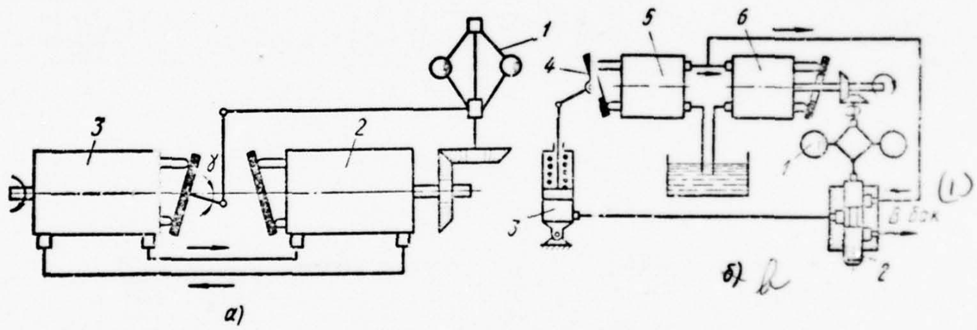


Fig. 180. Hydraulic drive of constant velocity with ball governor.

End Caption

Key: (1). Into tank. When using the uncontrolled hydraulic motors (with constant working volume) the pressure differential directly proportional to the torsional moment on the output shaft, which

appears under the action of the applied to it load, velocity on output shaft is proportional to the feed of pump.

During the application/use of a pump and motor with the adjustable working volumes (see Fig. 178c it is expanded the range of control, and also the possibility of the selection of output characteristics of drive on the torsional moments, powers and exit velocities. Calculated torque/moment and power in the interval of the frequency of rotation/revolution a of hydraulic motor correspond to the regulating of pump displacement, also, in interval of b - to the regulating of the working volume of motor. If the exit velocity of drive is regulated by a change in the controlled parameters of pump and motor, then theoretically similar drive will have the unlimited frequency band of the rotation/revolution - from infinitesimal to infinite according to expression (63).

However, if the output shaft of motor is loaded, then control can be carried out only within certain limits of the exit velocities, outside which angular velocity will not vary in proportion to to a change in the controlled parameter. The minimum working volume of motor $q_м$ will be the value, at which the developed with it torsional moment is capable to overcome both frictional resistance in motor and

the load, applied to its shaft.

In equal measure with some small pump displacements $q_н$ the rotation/revolution of the shaft of motor can cease itself due to the overflowing of liquid from the working cavities of pump and motor into nonoperative, and also due to its leakages into reservoir. This will begin at this value of the controlled parameter of the pump, with which the volume, described by its working cell/elements $q_н$, in one revolution will be equal to hydraulic slipes at this pressure or otherwise, when the volumetric efficiency of pump is equal to zero.

Hydraulic drive of the rotary action of constant velocity.

The examined drive of rotary action can be also applied and in such cases when necessary to automatically ensure the constant frequency of rotation/revolution in its output shaft at the sweep frequency of the rotation/revolution of pump spindle. The similar cases, for example, include the bringing into the rotation/revolution of the a-c generators of the aircraft, frequency of the rotation/revolution of engines of which they can change over a wide range (approximately 4:1) depending on flight conditions.

The schematic diagram of standard hydraulic drive of constant velocity with static centrifugal feed regulator is given in Fig. 180a. Drive consists of pump 3, adjusted on aircraft engine, and hydraulic motor 2, output shaft of which, connected with load, gives to rotation/revolution ball governor. Page 223.

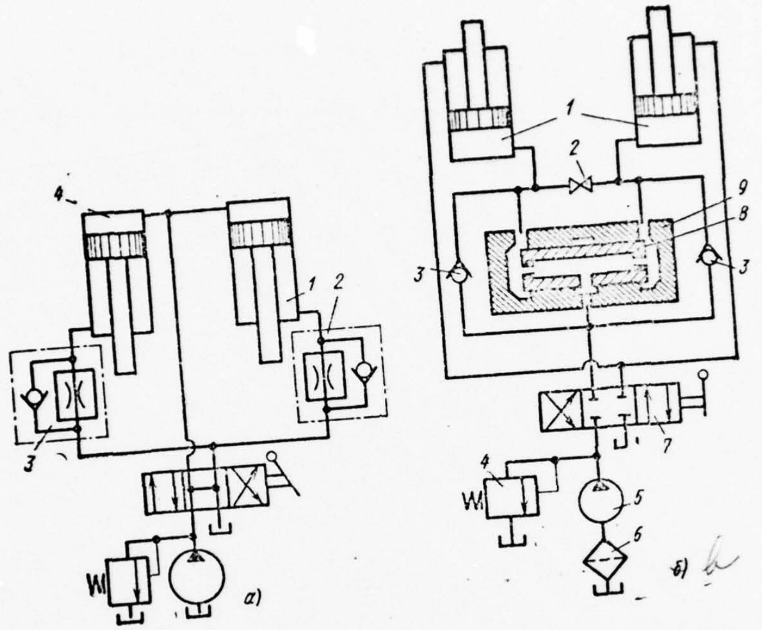


Fig. 181. [See following page for caption.]

Fig. 181. Systems with the synchronous motion of two actuating cylinders. End of Caption During a change (disagreement/mismatch) in the exit velocity of hydraulic motor connected with it ball governor 1 (Fig. 180a and b) affects in one manner or the other the inclined washer of pump and it changes angle γ its slope/inclination, and consequently, it changes the feed of pump. Virtually in the majority of constructions ball governor moves during a change in the frequency of the rotation/revolution of motor 6 (Fig. 180b) the plunger of the distribution valve of 2 control systems (Fig. 180a and b), which, supplying liquid to one cavity or the other of the hydraulic motor of 3 control mechanisms (angle of the slope of washer 4) of pump 5 (Fig. 180b), reduces disagreement/mismatch, supporting the exit frequency of the rotation/revolution of the hydraulic motor of constant.

Provision for synchronism and sequence of the motion of hydraulic engines.

In practice frequently it is required to ensure the synchronous motion of two and more actuating mechanisms.

The synchronization of the motion of output/yield of two and

more cylinders is realized in the majority of cases with the aid of the examined above throttle governors (limiters) of expenditure/consumption. The schematic diagram of a similar system is given in Fig. 181a. By the control of limiters (throttle governors) 2 and 3 are represented possible to ensure approximate equality or the assigned relationship of the velocities of the stock/rods of synchronized cylinders 1 and 4.

With the back stroke of cylinders liquid it passes through the check valves, established/installed in the housing of throttle governors 2 and 3.

Figure 181b depicts the diagram of system with the division of flow with the aid of regulator 9 (divider/denominator of the flow), in which the throttle/chokes are arranged in movable choke plunger 8. System is equipped by uncontrolled pump 5 with filter 6 and safety valve 4. Control is realized by three-position fourway distributor 7 (see Fig. 81d).

Back stroke (piston stroke of cylinders 1 down) is not regulated, for which system is equipped by check valves 3, through

which the fluid flow, displaced from cylinders 1, is headed for the circuit/bypass of regulator (divider/denominator of flow). For the disconnection/cutoff of regulator is applied pet cock 2. Page 224.

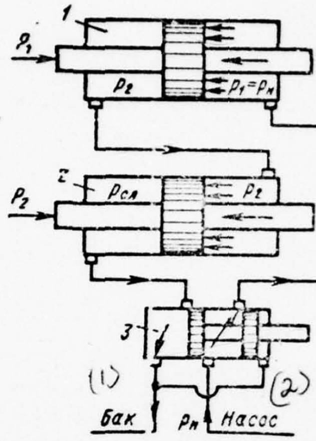


Fig. 182.

[See following page.]

Fig. 182. Systems with the provision for synchronous motion of two cylinders by means of series connection.

End of Caption.

Key: (1). Tank. (2). Pump. Are applied also the synchronizing circuits of two and more cylinders by means of their consecutive start. Figure 182 depicts the schematic of a similar connection/compound of two cylinder 1, it enters the right cavity of cylinder 2 in the position of distributor 3, shown in Fig. 182.

Assuming that the hydraulic slipes are absent and liquid is not compressed, we will obtain under the condition of the equality of the clear areas of cylinders the equal speeds of the displacement/movement of pistons, determined from the expression

$$Q_н = v_1F_1 = v_2F_2,$$

where $Q_н$ - the feed of pump; v_1 and v_2 are speeds of the displacement/movements of pistons; F_1 and F_2 are sectional areas of cylinders.

Since $F_1 = F_2$ we have

$$v_1 = \frac{Q_н}{F_1} = v_2.$$

The velocities v_1 and v_2 with this feed $Q_н$ are determined in this diagram by the working sectional area of one cylinder; however, the pressure $p_н$ pump in this case is determined by the total load of both cylinders $(P_1 + P_2)$, i.e., the pressure differentials for each of consecutive equipment/devices are summarized.

By disregarding friction of pistons and by counting drainage pressure $p_c = 0$, we will obtain for the two-cyclinder system

$$P_1 = (p_1 - p_2) F_1; \quad P_2 = p_2 F_2.$$

From these expressions it follows

$$p_1F_1 = P_1 + P_2;\ p_1 = p_{н} = \frac{P_1 + P_2}{F_1}.$$

Is common also the method of providing a sequence of the action of hydraulic engines with the application/use of diagrams with the matching hydroes-apparatus, which wear/operate at the signals of pressure. These apparatuses were called the name the valves of sequence and they are the distributive hydroes-apparatus, intended for the transmission of the flow of working fluid upon reaching of the assigned pressure in the supplying hydro-line. A similar diagram

of the hydraulic system, in which is provided the assigned sequence of the action of two actuating cylinders (hydraulic engines), reached by special matching valve 3 of series connection, is represented in Fig. 183a. The supply of liquid into the first cylinder of 1 this diagram conducts directly from distributor (in the diagram is not shown), but in second cylinder 2 - through matching valve 3. After the pressure in cylinder 1 will be raised at the termination of working piston stroke to the value, at which are adjusted the spring of matching valve 3, its plunger will move and will discover channel a of the power supply of cylinder 2.

Figure 183b gives the diagram of the hydraulic system of a similar designation/purpose with throttle/choke 3, established/installed in the feed line cf first cylinder 1. The sequence of the action of actuating cylinders 1 and 2 in this diagram is provided by the fact that in feeder is supported because of the presence of throttle/choke 3 jump/drop in the pressure of liquid, under action of which the gate of matching valve 4 is held in the right position, in which it overlaps the feed line of second cylinder 2. Page 225.

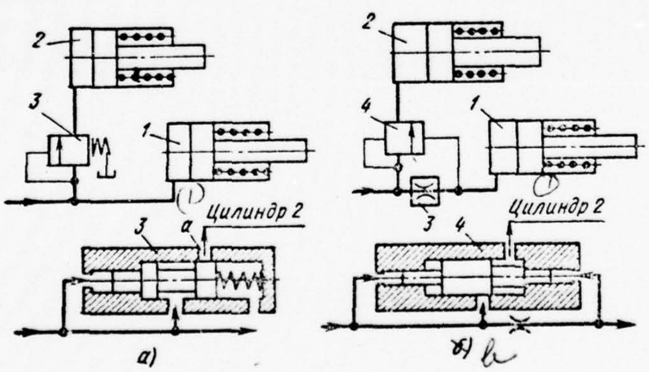


Fig. 183. Schematics of the provision for sequential action of two actuating cylinders.
Key: (1) Cylinder.

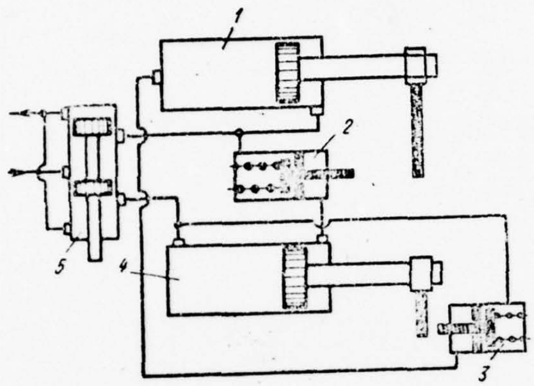


Fig. 184. Schematic of the provision for sequential action of two cylinders in both directions.

After the pressure differential on throttle/choke 3 as a result of a pressure increase in first cylinder 1 decreases, the gate of valve 4 under the action of the increasing pressure will discover channel for the duct of liquid in second cylinder 2.

Are applied the diagrams, in which the sequence of action of actuating cylinders is provided mechanically with the aid of the pushers, which affect the changeover valves. The similar hydro-diagram, in which the sequence of motion is provided both on straight line and on back the strokes of actuating cylinders 1 and 4, is depicted on Fig. 184. In the position of distributor 5, presented in the diagram, first is accomplished the working stroke of first cylinder 1 and only after at the end of its course by detent on stock/rod will be discovered matching valve 2, into motion comes the piston of second cylinder 4.

During a change in the direction of fluid flow back stroke will accomplish first second cylinder 4 and only at the end of its course by detent it will be include/connected second matching valve 3, which includes feed in first cylinder 1.

In many instances it is required tc arrange distributors in such a sequence in order that one any put into action distributor would overlap fluid flow in all others, arrange/located after it, i.e., in crder to exclude the simultaneous work of several distributors and actuating mechanisms.

In such systems the outlet duct of the first valve is connected with the inlet duct of the second and so on. The diagram of a similar system is given in Fig. 185.

The cavities of the supplied actuating cylinders (hydraulic engines) in the mid-position of distributors can be either blocked (Fig. 185a), or connected between themselves and the tank (floating position), as shown in Fig. 185b. Page 226.

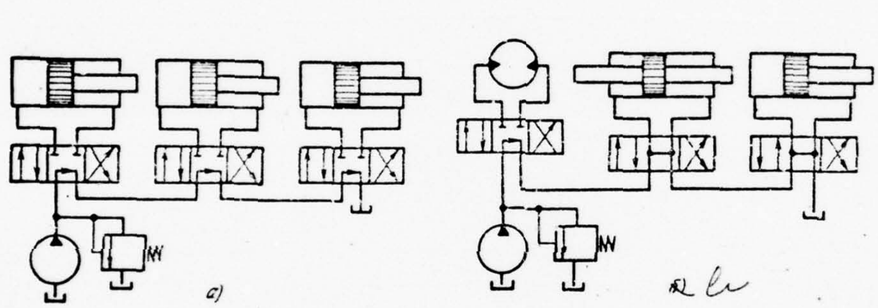


Fig. 185. Series circuits of hydraulic engines. End of caption In the latter equipment/device with actuating cylinder simultaneously can work only one hydraulic engine, and only in the system, in which are applied one or several rotor continuous hydraulic motors, simultaneously they can work several hydraulic engines (Fig. 185b). The drainage pressure of this engine can be used as operating pressure of the subsequent engine. The pressure differentials along an entire series acting similarly of equipment/devices store/add up.

Heat balance of hydraulic system.

Since the energy, lost in hydraulic system, is converted into heat, the temperature of liquid is raised and under known conditions can achieve inadmissible values. During an increase in the temperature is reduced the viscosity of oil (see page 15), which is accompanied by an increase in the leakages and by the progressive increase in the temperature because of energy loss at leakages. Besides this the elevated temperatures are undesirable also as a result of an increase in this case in the oxidation of oil, which is accompanied by further fall in the viscosity and by the formation/education of resins and different residue/settlings (see page 17).

By taking into account this, it is necessary that in the hydraulic system be established/installed the corresponding heat balànce, determined by the equality of inflow and diversion/tap (evacuation) of heat. When the temperature of oil exceeds permissible, one should apply air or water cooling. The virtually

acceptable temperature of oil in hydraulic system is 50-60°C.

Approximately it is possible to count that into system enters per unit time the amount of heat, to an equivalent difference in the complete (driving/homing) lifting power and net power (effective shaft horsepower of hydraulic motor or on the stock/rod of actuating cylinder). If during some interval of time useful work does not conduct, then into heat is converted per unit time entire/all driving/homing lifting power.

In accordance with that which was indicated the lost in hydraulic system power will be determined from the expression

$$N_{nom} = N_{no\partial}(\quad -\eta),$$

where $N_{no\partial}$ - power input (driving/homing lifting power); η - complete plant efficiency (system).

To power N_{nom} is equivalent the flow of heat (amount of heat per unit time)

$$A = N_{nom}k = N_{nc\,)}(1-\eta)\,k,$$

where k - the heat equivalent of mechanical energy (for a power 1 kW it is equal to 860 kcal/h, for a power 1 h.p. - 632.4 kcal/h). In unity of SI k = 1; if N is expressed in watts, then unity of the flow of heat A is expressed watt (W).

In many instances (with throttle control) the utilized by users hydraulic power is virtually close to zero, and consequently, entire/all work of hydraulic setting up is converted into heat. Page 227.

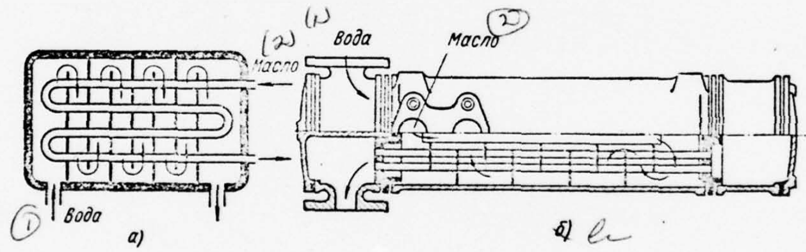


Fig. 186. Water coolants of the liquid of hydraulic systems.

Key: (1). Water. (2). Oil. An increase in the temperature of liquid during its extrusion through choke slots can be approximately determined, after equating the energy, given up by the escape/ensuing from slot liquid in volume of V, to the energy, expended for the heating of the liquid of this volume (not taking into account that part of the isolated heat departs from system as a result of heat

emission):

$$V\,\Delta p = V\rho cm\Delta t, \tag{65}$$

where V is a volume of the liquid, which takes place through the slot; Δp - the loss (jump/drop) of pressure in slot; ρ is density of liquid (for mineral oils ρ = 900 kg/m³). c is liquid specific heat [for oils c = 0.45 kcal/ [kg•°K)]; m - the mechanical equivalent of heat (m = 427 kg•m/kcal; in SI m = 1); $\Delta t = t - t_0$ is an increase in the temperature of liquid; here t and t_0 is the unknown and initial temperature of liquid.

Increase in the temperature

$$\Delta t = t - t_0 = \frac{\Delta p}{\rho cm}.$$

After accepting for widespread mineral oils ρ = 900 kgf/cm³ and c = 0.45 kcal/ (kg•°C), expression (65) can be reduced to the form

$$\Delta t = 0{,}058\ \Delta p,$$

where Δp is the pressure differential in kgf/cm^2.

From this expression it follows that with the throttling/choking of pressure oil from 100 kgf/cm^2 to the atmospheric its temperature is raised in one operation/pass through the throttle/choke approximately on 6 °C.

During the known mode/conditions of the work of hydraulic system it will be required to use for providing the assigned temperature conditions the coolers, which are applied, as a rule, in the continuous operation of hydro-electric power plants with power 20-30 h.p., although in a number of cases these equipment/devices are applied in smaller powers.

In the majority of the coolers (heat exchangers) of hydraulic systems as the cooling medium are utilized the water or the air, although are applied also other liquids. For example, in aircraft as the cooling medium is utilized fuel/propellant.

Figure 186a depicts the diagram of the simplest water-butyric radiator, which is the placed into water tank coil, on which is passed oil. Tank (jacket of coolant) is equipped by the system of the partition/baffles, soldered to duct, which create the circulation of water and the intense diversion/tap of heat.

Frequently the heat exchanger in the form of the coil, on which is passed the cooling water, is incorporated into oil tank of the hydraulic system.

Page 228.

Are common also the tubular (honeycomb) coolers (heat exchangers), which consist of assembled into pencil of straight lines ducts (hundreds), included into housing (Fig. 186b).

Heat exchangers establish/install, as a rule, in drain line. It

is necessary to exclude the possible pressure overshoots in this main line, which can break heat exchangers.

LIQUID MAIN LINES AND TANKS.

The supply of liquid under operating pressure from the source of expenditure/consumption (pump) to the users of hydraulic power (to performing hydraulic engines) and its diversion/tap from the latter under drainage pressure conducts through the liquid main lines, as which serve the rigid and elastic metallic and nonmetallic conduit/manifolds, and also the channels, executed in the housing of hydroaggregates.

It is accepted to call the section of conduit/manifold, which connects pump 1 with reservoir (tank) 5, suction line (see Fig. 3a); the section of the conduit/manifold, on which the liquid from pump 1 enters hydraulic engine 2 - by pressure (worker or pressure) main line, and the section of the conduit/manifold, through which the liquid is abstract/removed from the nonoperative cavity of hydraulic engine 2 into reservoir 5, by drain line. The force main includes also all main lines, under operating pressure.

Calculation of the section of conduit/manifolds.

The calculation of the section of conduit/manifolds (channels) is performed on the basis of the mass or volumetric fluid flow rate through the clear opening of channel. Accordingly, the fluid flow rate through this conduit/manifold is determined by area f of its internal section and by average speed u of the flow of liquid, which are connected by the relationship

$$Q = fu; \quad m = fu\rho,$$

where Q and m - volumetric and mass fluid flow rate; ρ is density of liquid.

The calculation of the section of other channels of hydroaggregates, along which flows the liquid, conducts on the basis of the law of flow continuity (constancy of the expenditure/consumption), according to which the expenditures in the different sections of flow during steady motion are identical:

$$u_1f_1 = u_2f_2,$$

or

$$\frac{u_1}{u_2} = \frac{f_2}{f_1},$$

where u_1 and u_2 are identical the average rates of flow of liquid in sections f_1 and f_2 of channels.

When selecting the rate of flow of liquid in conduit/manifold they are guided by the fact that an increase in it leads to an increase in the resistance and with respect to power loss, but reduction - to an increase in the mass of conduit/manifolds and accessories, and also to the adverse structural solutions of the cell/elements of hydraulic drive.

The loss of pressure in conduit/manifold, which is the measure of its hydraulic efficiency, is caused by the resistance of the quite straight portion of conduit/manifold as well as by any local /

resistances: by the changes in the sections of conduit/manifolds and by its curvatures, and also the presence of different hydroaggregates.

On the basis of practice it is possible to recommend for the conduit/manifolds of force mains the following rates of flow of liquid in pressure dependence:

pressure in kg/cm²	50	60	150	200
velocity in m/s.	3,0	4,5	5,5	6,0

Page 229.

For inlet tubing of self-primings pump the velocity of liquid is selected by 0.5-1.5 m/s, the smaller values accepting for the conduit/manifolds of a small (10-20 mm) section. The velocity in the conduit/manifolds of drain lines usually they restrict 2-2.5 m/s.

However, in many instances, and in particular with the application/use of low-viscosity liquids of the type of AMG-10 and at positive ambient temperatures, are accepted the velocities:

5-10 m/s - for pressure and drain lines;

2-3 m/s - for suction lines.

Sometimes, in particular at high operating pressures, are applied the velocities to 30 m/s.

As the conduit/manifolds of hydraulic systems are applied jointless steel cylindrical pipes and thinner than the duct from nonferrous metals and different plastics.

The calculation for the longitudinal discontinuity of the direct/straight thin-walled conduit/manifolds, loaded with the internal static pressure, at which it is possible to disregard the secondary stresses, which appear as a result of the distortions of

the cylindricity (ovality) of the section of conduit/manifold, can be produced on the formula

$$\sigma_p = \frac{pd}{2s}, \qquad (66)$$

where σ_p - the allowable voltage of the material of the ducts of wire with the elongation (in circumference), which usually is selected equal to 30-35o/o of time/temporary strength of materials of conduit/manifold; p is the maximum pressure of liquid in kgf/cm²; d and s is a outside diameter and the wall thickness of duct in cm.

By thin-walled are understood the conduit/manifolds, in which the ratio of the outside diameter d to thickness s of its wall satisfies the condition

$$i = \frac{d}{s} \geqslant 16 \text{ or } \frac{d}{d_{\text{вн}}} < 1{,}7,$$

where $d_{\text{вн}}$ is an inner diameter of the section of conduit/manifold.

For conduit/manifolds with this ratio of diameter to wall thickness (more than 16:1) they assume that the outer diameter is equal internal.

During calculations of steel conduit/manifolds the tensile strength in kgf/cm² can be accepted according to the given below data:

the stainless steel	C20	30ХГСА
3500	4100	12000

Wall thickness taking into account the probable deviation of diameter and of wall thickness calculate according to the expression

$$s = \frac{p(d + m)}{2\sigma_p n},$$

where m = 0.3 - deviation with respect to the diameter of conduit/manifold in mm (GOST - All-union State Standard] 8734-58); n = 0.9 - the coefficient, which considers deviation with respect to the wall thickness of conduit/manifold (GOST 8734-58).

For the calculation of thick-walled conduit/manifoli (i > 16), in which the voltage/stress changes from the maximum value on internal wall to the minimum on external wall, applies Lame's formula:

$$\sigma_p = p\frac{d^2 + 2s + 2s^2}{2s(d - s)}. \quad (67)$$

Minimum wall thickness for this conduit/manifold

$$s_{\min} = \frac{d}{2}\left(\sqrt{\frac{\sigma_p + p}{\sigma_p - p}} - 1\right).$$

Page 230.

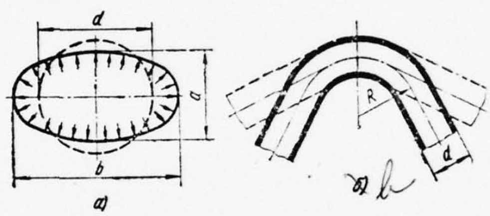


Fig. 187. Deformation of conduit/manifold under the effect of pressure of the liquid: a) transverse; b) axial. Fatigue strength of conduit/manifolds.

The conduit/manifolds of many machines undergo the simultaneous loads of static and dynamic character. The first include the examined static loads, caused by the internal pressure of liquid, and the loads, which develop during the assembly of conduit/manifold, and also the load, which appear as a result of the thermal deformations of conduit/manifolds and cell/elements of the machine, to which are

fastened the conduit/manifolds. To the second are related the loads, appearing during the frequency deformations (oscillations) of conduit/manifold, caused by the pulsation of the pressure of liquid, by hydraulic impacts, and also the oscillations of conduit/manifolds themselves, caused by internal and external disturbance/perturbations. The voltage/stresses, which appear in the material of conduit/manifold, represent the sum of the enumerated components.

Significant of the dynamic loads are the loads, caused by fluctuations (pulsation) of the pressure of liquid. These fluctuations of pressure are caused by kinematics and the special feature/peculiarities of the mode/conditions of the work of pumps, and also by the hydraulic impacts, which are observed during the instantaneous function of different valves.

During the agreement of pulse frequencies of pressure and fluctuations of liquid column in conduit/manifold in the latter can arise the internal resonance, in the presence of which the amplitude of fluctuations of pressure considerably will increase.

Frequency ω the natural oscillations of liquid column, which fills the cut of duct with a length of L, in the general case can be determined by expression [7, 14]

$$\omega = \frac{a}{L}$$

where a is the speed of sound in the liquid, which fills conduit/manifold; L = a/f - the wavelength of the fluctuating pressure; f is frequency of disturbance/perturbations (pressure impulses).

Observations show that the considerable number of cases of the fatigue failures of conduit/manifolds, and in particular during the pulsations of pressure, is caused by the disturbance/breakdown of the cylindricity of the cross section (by presence of ovality) of the latter (Fig. 187a). In this case unlike the conduit/manifold of the round cross-section whose deformation occurs only because of lengthening the perimeter of its section, oval section approaches under the action of the internal pressure of liquid the round cross-section with a diameter of d, although not all points of

perimeter strictly follows this law. In view of this at the points of the greatest curvature oval section appear the high voltages, which depend on the flatness (ovality) of cross section and are characterized by the relation

$$k=\frac{b-a}{b},$$

where a and b are size/dimensions of small and major axes of oval.

On the basis of test data, and also experience established/installed that the maximumly permissible ovality for the steel conduit/manifolds of the widespread size/dimension is k = b - a / b•100 = (4-5) o/o.

Page 231.

The strength of conduit/manifolds affects a radius of its curvature. The bent conduit/manifold under the action of the forces

of pressure of liquid attempts to straighten (Fig. 187b), as a result of which its in place maximum curvature can arise the considerable stresses, which lead during frequency fluctuations of pressure to fatigue failures. Since in the zone of the maximum curvature usually occurs the maximum flatness (ovality) of the section of conduit/manifold, this zone it is the most probable place of destruction.

According to experimental data, on the sections of main line, which work under conditions of the high fluctuating pressures, one should apply bending radii R > 3d, where d and R are the outer diameter of conduit/manifold and the bending radius of its axle/axis.

In the general case the allowable voltage, according to expressions (66) and (67), for the conduit/manifolds, which work under conditions of fatigue loadings by the fluctuating pressure with the amplitude of pulsation, by approximately equal to 40-50o/o of operating pressure, must be lowered approximately 2 times in comparison with allowable voltage for the conduit/manifolds, which work under conditions of static loading.

Resonance oscillations of conduit/manifolds.

Resonance oscillations can arise as a result of vibrations and relative displacement of machine parts, to which are fastened the conduit/manifolds, and also as a result of effect on the pipeline of the examined above fluctuating forces of pressure of liquid. If one end of the pipeline is oscillate as a result of the vibration of machine parts relative to another with the frequency, equal to the natural vibration frequency of the section of pipeline in question, then pipeline can enter the resonance oscillations, during which the amplitude of the oscillations of the middle part of the pipeline can into dozens and more once exceed the amplitude of the perturbation (exciting) oscillations of the ends of the pipeline.

The possibility of the emergence of the flexural resonance oscillations of the bent pipeline is caused also by the fact that the pipeline will attempt under the effect of pressure of liquid to straighten (see Fig. 187b), as a result of which at the fluctuating pressure liquid the bent section of pipeline it can enter the flexural vibrations.

The natural vibration frequency of any section of pipeline depends on a number of factors, and in particular on the character of the stopping up of its ends.

With the rigid stopping up of both ends, what corresponds to the widespread in practice method of the attachment of pipelines, natural vibration frequency in Hz of rectilinear pipeline can be determined taking into account the weight of its filling liquid by the empirical expression

$$\omega=\frac{3{,}56}{L^2}\sqrt{\frac{EJg}{G_T+G_{ж}}},$$

where L - distance between supports; E is modulus of elasticity of material; J - the second moment of area of duct; G_T and $G_{ж}$ - the weight per unit length of pipeline and liquid.

The natural vibration frequencies of pipeline it depends on internal pressure and the rate of flow of liquid. Taking into account the effect of these frequency factors of natural oscillations in Hz

$$\omega' = \omega \sqrt{1 - \frac{P}{P_{\kappa p}}},$$

where $P = pf + m\,u^2/2$ - the pressure of liquid in pipeline; f is an area of the internal section of pipeline; m - linear density, i.e., the mass of the unit of length; u - the rate of flow of liquid in pipeline; $P_{\kappa p} = \frac{\pi^2 EJ}{L^2}$ is critical force according to Euler. Page 232.

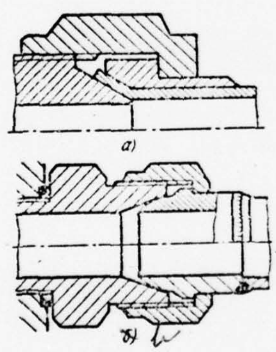


Fig. 188. Types of union couplings: a) by rolling-out; b) with the aid of nipple. Union couplings.

For the connection/compounds of thin-walled tubes (to 30-35 mm)

in essence is applied the accessories under the rolling-out of ducts on outside cone (Fig. 188a), which in this case must be made from the ductile metal, which allow/assumes rolling-out in cold state. Are common the angles of rolling-out from 30 to 90° (in the USSR - 60°, in England - 30° and in the USA - 37°).

Connection/compound with the rolling-out of duct differs in terms of simplicity, but it can be recommended for steel tubes only at pressure 200-300 kgf/cm².

For higher pressures (300-400 kgf/cm²) apply nipple (spherical) connection/compound (Fig. 188b). The airtightness of this connection/compound is provided by the contact of the surface of steel spherical nipple with the conical surface of branch.

Are applied also other union couplings [5].

Flexible conduits.

When occurs the displacement of twc machine parts, to which are fastened the ends of the pipeline, are applied the connection/compounds, which allow/assume similar displacements. Similar connection/compounds include the flexible conduits - the rubber-fabric hoses, intensified by metallic cover/braid, and flexible metal hoses.

Since the basis/base of flexible hose in the majority of cases is rubber, they are suitable only for temperatures to 35°C. For a work under conditions of high and low temperatures are applied the flexible conduits (hose/pipes) with the metallic bellows, with parallel (Fig. 189a) either spiral (Fig. 189b) corrugations, the included for increase in strength and protection from mechanical damages into one or multilayer wire cover/braid (Fig. 189c).

Metallic hose/pipes manufacture for works under conditions of temperatures from -100 that 540°C and for operating pressures at small diameters of section (about 6 mm) to 400 kgf/cm².

Pulsations of pressure and loss of head in hose/pipes.

The corrugated form of flow area of flexible metal hose causes the appearance of pulsations, and also it causes supplementary losses of head. The first are caused by the fact that on hose/pipe the fluid flow taking place experience/tests pericdic expansions and the compression, caused by the undulation of flow area of the channel (hose/pipe). The frequency f of these effects is determined by the amount of corrugations and by the time τ course through the hose/pipe of the liquid:

$$f=\frac{l}{t\tau},$$

where l it is determined the length of the flexible part of the hose/pipe; t is a space of corrugation.

After expressing time τ by velocity u_{cp} flow of liquid

$$\tau=\frac{l}{u_{cp}},$$

we will obtain

$$f=\frac{u_{cp}}{t},$$

whence it follows that the ripple frequency increases with an increase in the velocity u_{cp} flow of liquid and with a decrease in the space of corrugations t. Page 233.

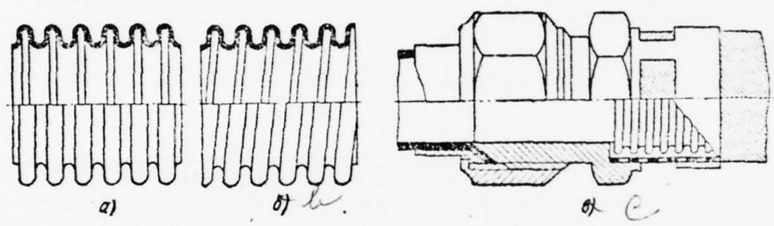


Fig. 189. Flexible metal hose. End of caption. The pulsation of liquid leads under specific conditions to the appearance of the longitudinal resonance. The observations showed that during approach/approximation to certain fluid flow rate (with an increase u_{cp}) appears the sound of high tone, and the temperature of hose/pipe at a distance 10-20 mm from exit nipple sharply it is raised. After 2-3 min under these conditions the hose/pipe is destroyed, whereupon the investigation of the places of damage reveal/detected the circular cracks of fatigue character.

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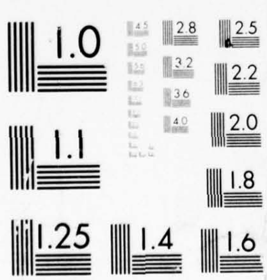

MICROCOPY RESOLUTION TEST CHART
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this hose/pipe depends on the geometric coefficient of $m = zd$, where z depends the turn number or corrugaticns on 1 cm. of length and d - the inner diameter of hose/pipe.

For $Re \geqslant 1800$ coefficient λ in formula (68) can be determined by the empirical equation

$$\lambda = 0{,}4\left(\frac{h}{d}\right)^{1{,}6} zd.$$

For calculations Re it is accepted that d - the diameter of the internal part of the corrugation.

For the approximate computations of losses of head in hose/pipes during turbulent mode/conditions (Re > 2000) it is possible also to use expression (14), whereupon coefficient λ for this case is calculated from the formula

The calculation of losses of pressure is performed in the general case according to expression (14):

$$\Delta p = \lambda \frac{L}{d} \cdot \frac{u_{cp}^2}{2} \rho. \quad (68)$$

Critical Re, which corresponds to transition from laminar flow conditions to turbulent mode/conditions, for these hose/pipes is equal to $Re = \frac{u_{cp} d}{\nu} = 1800 - 2000$. The average speed u_{cp} flow is accepted on the basis of the conditional diameter d of corrugations in world/light.

In the range of turbulent flow conditions λ it does not depend on Re, but it is determined by the relative undulation of the internal surface of the flexible part of the hose/pipe, expressed by ratio h/d, where h is determined the middle interior height of corrugation and d - the inner diameter of hose/pipe (on the internal apex/vertexes of corrugation). Furthermore, the hydraulic friction of

$$\lambda = 0{,}6\mathrm{Re}^{-0{,}25}.$$

Consequently, drag coefficient λ for hose/pipes approximately 2 times is higher than the drag coefficient for smooth-walled ducts.

Reservoirs (tanks) for a liquid.

By tank is understood the storage/accumulator of working fluid, which is located by atmospheric or overpressure, which can simultaneously obtain working fluid from drainage hydro-line and give up it in the suction hydro-line. The minimum capacity of tank is determined in essence by the change of the capacity of the aggregate/units of hydraulic system, which occurs in the process of work. Page 234.

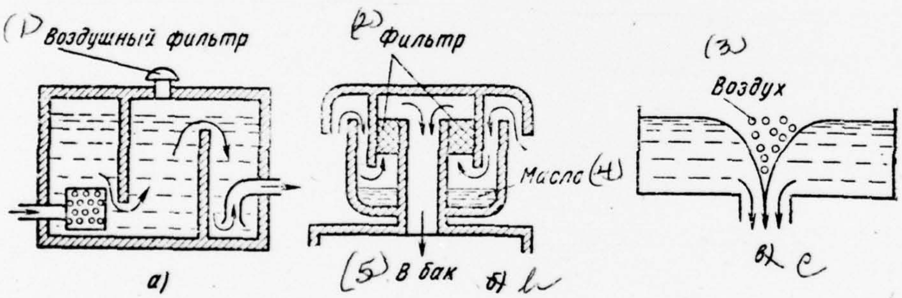

Fig. 190. Diagrams of tanks for working fluids.

Key: (1). Air filter. (2). Filter. (3). Air. (4). Oils. (5). Into tank. End of caption. Otherwise, the capacity of tank must be a sufficient for an insertion entire working fluid, which pours from system, and maintaining the necessary level of liquid during the execution of operating cycle. Virtually the capacity of tank usually is selected to the equal 2-3- minute feed of pump.

Tank must be designed so that in it would be provided residue of the liquid. For this the circulation of liquid in reservoirs must be brought to the minimum. The input/introduction of liquid into tank must not cause frothing and its turbulence. For this it must be arranged below liquid level in tank. On lead-in channel (duct) one should establish/install mesh equipment/device (Fig. 190a) for the jet disintegration of liquid. For the elimination of the incidence/impingement into the tank of dust together with the air, which enters it during changes in liquid level, drain hole must be equipped by the air filters, which recover dust and moisture. The more advanceed method of the preservation of tank from the incidence/impingement of dust is the application/use along with the filters of liquid dust catcher (Fig. 190b).

To drain lines it should be connected to the special sections of tank, moved away from the sections, connected with suction lines. This is provided by the application/use of partitions (Fig. 190a) with height/altitude, the equal to 2/3 height/altitudes of the minimum oil level in tank.

It is necessary also to follow the preservation/retention/maintaining of the required level of liquid in tank, since a fall in it will cause its intense circulation in tank, which can lead to the incidence/impingement of air into liquid from without.

With fall in the tank of liquid level in the places of the connection of inlet tubing can be formed the funnel (Fig. 190c), through which the air will fall into pump. The formation/education of funnel is caused by eddy/vortices, and also fact that at certain height/altitude h of liquid column under intake opening the hydrostatic pressure $p = \gamma h$ this opening/aperture, created by gravity strength of liquid column, becomes so/such small, that does not provide the horizontal displacement of the layers of liquid to the axle/axis of the opening/aperture, required for the completion of the feed of pump.

Pages 235-256.

Chapter VI.

PACKING/[redacted] OF THE CONNECTIONs/[redacted] OF HYDRAULIC SYSTEM.

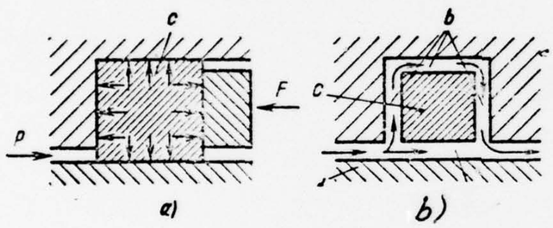

Fig. 191.

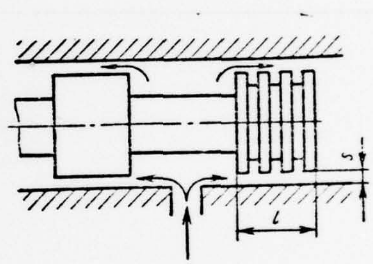

Fig. 192.

Fig. 191. Diagrams, which illustrate the operating principle of the sealing cell/elements.

Fig. 192. Diagram of the noncontact sealing/pressurization of plunger. ¶ Airtightness is a property of construction not to pass through itself working medium in the amounts, which exceed maximum permissible. For this are applied the different sealing means whose

designation/purpose lies in the fact that, in order to impede hydraulic slip, which is located under certain overpressure, through the gap clearance of two motionless or being moved one relative to another rigid surfaces, components the unit. The latter is achieved by the elimination of the clearance between the packed surfaces with the aid of any soft elastic material, placed between them (bearing seal), which is shown in Fig. 191, or by the provision for the small clearance s between the surfaces of the combinable parts (for example, noncontact packing/seal), which is shown in Fig. 192.

Closed-butt joint is achieved by any soft elastic material, placed between the packed surfaces (Fig. 191a), which under the action of the external force of F or forces of pressure of liquid is adjusted to these surfaces, creating close contact.

Figure 191b are schematically shows the possible channels of leakages in the node/unit of the packing/seal of the movable stock/rod d, which must be tightly overlapped by the soft sealing cell/element c. It is obvious that greatest difficulty will present the overlap of channel a, i.e., the interface pressure integrity of movable connection, in view of which to accuracy/precision and surface finish characteristics, which form this channel, are

presented especially high requirements. The sealing/pressurization (overlap) of channels b, formed by soft sealing cell/element and the motionless surfaces of node/unit, is provided considerably simpler, since here virtually will be packed motionless compound. The leakages, caused by the permeability (leakage/looseness) of the material of sealing cell/element, are removed by application/use for its production of the materials of the corresponding densities, as which in essence are utilized rubber and resin-like materials. Page 236.

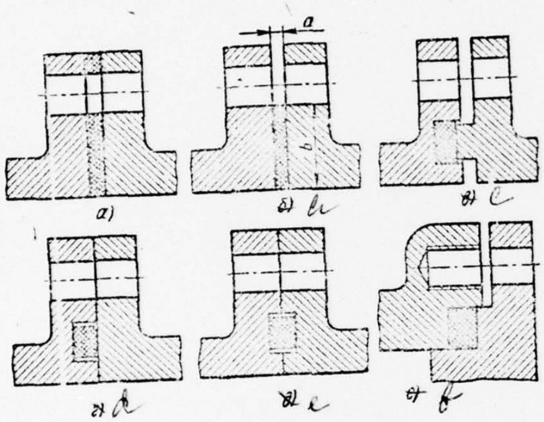

Fig. 193. Sealing/pressurization of the motionless compounds of joints.

The process of sealing/pressurization by rubber parts is realized because of the introduction of compressed rubber in the irregularity of the contacting with it surfaces; in this case occurs the filling with rubber macro- and the microchannels of the packed surface and their overlap.

STATIC SEAL.

For the elimination of hydraulic slipes through the motionless compound they attain elimination by the different means (predominantly pads) for the clearance between mating parts, whereupon hermetic seal will be reached, if contact points form closed curve, and contact pressure by it exceeds the pressure of the packed medium. By contact pressure here is understood the external compressive force, per unit of the surface area of packing, which is developed with the tightening of the bolts of flange joint.

In some constructions the airtightness is provided by breaking in of parts; however, since to ensure this accuracy/precision, during which the contact points form closed curve, in this manner it is

difficult, it is applied only for the interior conjugations, in which the provision for a complete airtightness is not required (slide-valve vapors, the valves, etc.).

Some methods of static seal by pads and the rings represented in Fig. 193. For the production of packing apply the different nonmetallic and metallic elastic materials, able to compensate for in the tightening of the compound of inequality and other surface defects that which is packed the vapors.

Packing must be protected from extrusion, for which preferably place them into the grooves, forming closed cavities (Fig. 193c and f). In such a case, when these means are not provided, it is necessary that the frictional force of packing against contact surfaces be more than the force of pressure of liquid on its lateral surface, which is reached by the selection of thickness a and of width b of packing (Fig. 193a and b).

The ferrules with rectangular cross section, prepared from elastic material, are placed in grooves and are designed usually for their complete (with certain surplus) filling (Fig. 193c and f). For

this the cross section of groove is selected to 30o/o less than cross section of ferrule (packing).

When it is required to ensure precise axial location of parts of compound, and also necessary to unload packing of the effort/forces of the tightening of the bolts of compound, is applied the flange joint, shown in Fig. 193d and e. The volume of packing in this case must be somewhat less (to 10-15o/o) the volume of groove; however, its section in free state must be so that with assembly would occur the compression of rubber by height to 20-25o/c in comparison with size/dimension in free state (see also page 24). Page 237.

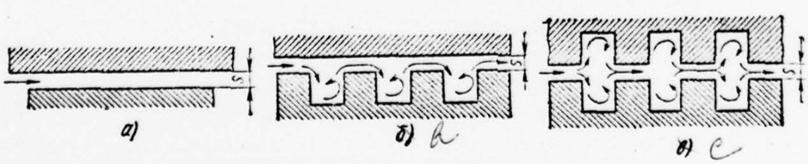

Fig. 194. Diagrams of the packing/seals: a) slotted; b and c) slotted and labyrinth.

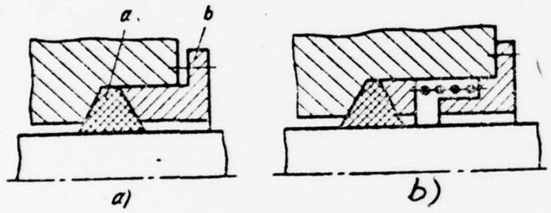

Fig. 195. Sealing/pressurization by packings. ¶ By the widespread type of static seal is also the packing/seal the rings of round cross-section.

DYNAMIC SEAL.

Dynamic seals it is possible to divide into two groups: noncontact (slotted) and contact.

By the first are understood the packing/seals, in which the required airtightness is provided by the hydraulic friction of the slot, formed by the surfaces of the packed pair (static packing/seals). This packing/seal, obtain the name slotted, is the capillary smooth slot s (Fig. 194a), with the appropriate value and at length of which can be created taking into account the obliteration of slot the acceptable friction to overflowing of liquid.

Similar noncontact compounds (packing/seal) are applied when to packing/seals do not present the requirements for the provision for a complete airtightness (for example, for a decrease in the overflowing of liquid of one cavity of hydroaggregate with high pressure in another cavity with smaller pressure).

For an increase in the friction of slot with high Re, which correspond to turbulent flow conditions (predominantly with gas working media), on one (Fig. 194b) or both (Fig. 194c) surfaces, which form slot, make the labyrinth grooves, which as a result of the alternating change in the section of slot raise under known

conditions its friction. Furthermore, the application/use of these grooves on the plungers of valves and valves contributes to their discharging from the unbalanced radial forces of pressure of liquid.

From bearing seals simplest are the stuffing box packings (Fig. 195) from any soft material a. During the compression of packing by the pressure bush b the packing flows in radial direction, forming the close contact between the chamber of gasket and the packing on the one hand and movable part (stock/rod or shaft) and by packing - on the other hand. For lightening the creation of the requiring contact of gasket with the packed movable part the faces of the chamber of gasket usually are made at an angle (conical shape).

For the compensation for wear and other losses of volume, the packing gaskets require periodic suspenders. This is achieved by the indicated compression of packing with the aid of the bolts (Fig. 195a) or a spring (Fig. 195b). These packing/seals are applied in the small pressures of the packed medium (to 50 kgf/cm^2). Page 238.

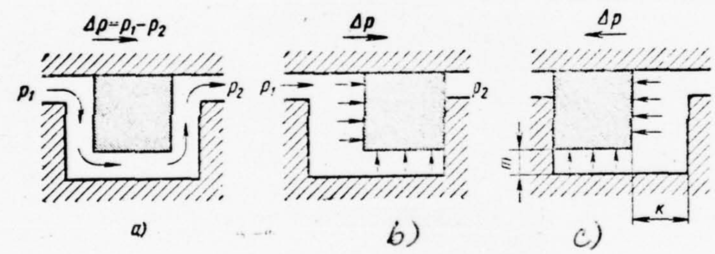

Fig. 196. Sealing/pressurization by sectional ferrules.

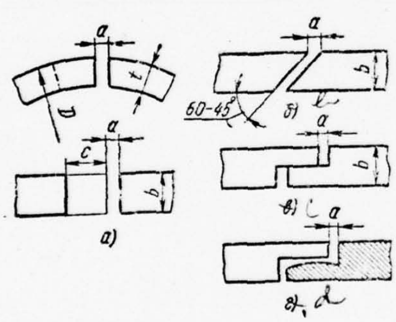

Fig. 197. Types of the butting locks of sealing ferrules.

Metallic piston rings. ¶ In hydroaggregates with rectilinear motion are common piston packings and plungers with the aid of elastic sectional piston metallic and plastic rings. The diagram of the action of packing/seal is shown in Fig. 196. Ring from the position, presented

in Fig. 196a, under effect of pressure of liquid is moved to the appropriate operating position (Fig. 196b and c). The sealing contact of ring with the surface (mirror) of cylinder is created by the spring action (radial elasticity) of ring, which develops with its assembling reduction, and also with the pressure of liquid on the lower surface (from the base of groove) of ring and in axial direction by the pressure of the packed liquid (Fig. 196b and c).

Experiments show that the rings (with two-three rings in sealing node/unit) in question provide high (virtually complete) airtightness at pressure more than 210 kgf/cm^2 and the during qualitative production.

Rings are manufactured from the material, which possesses sufficient elasticity and antifriction properties. Specifically, the rings are manufactured from gray cast iron, bronze, Textolite, graphite and metal-graphite mass.

Apply straight line (Fig. 197a), by scythe (Fig. 197b) and stepped (Fig. 197c) the joints (locks) of rings. Direct/straight joint applies in low pressures (to 50 kgf/cm^2), by scythe (angle 60°)

- in mean pressures (50-200 kgf/cm^2) and stepped - in higher pressures, and also in the increased requirements for airtightness. In stepped lock (see Fig. 197c) the being joined stepped ends of the ring cverlap each other, decreasing the butting clearance. Frequently one of mating surfaces in lock is made by plane (parallel face), and the second - several convex (see Fig. 197d), thanks to which is raised the specific pressure in the joint of rings under load, which facilitates an increase in the airtightness.

The value of the butting clearance c + a of ring (see Fig. 197a) in its free state and value c, by which this clearance decreases during the assembly of piston with ring to cylinder, define the ring strain both in the compressed position and with its putting on on piston. For practical calculations it is possible to accept c = 3.4t, where t is radial thickness (height/altitude) of the section of ring. Page 239.

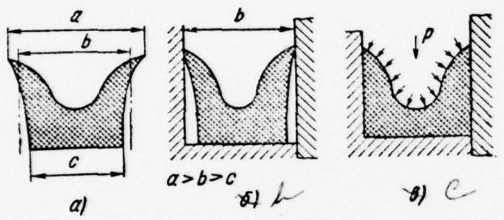

Fig. 198. Diagram of the action of the cup packing: a) collar to assembly; b) collar in the installed form without the pressure of liquid; c) collar under pressure.

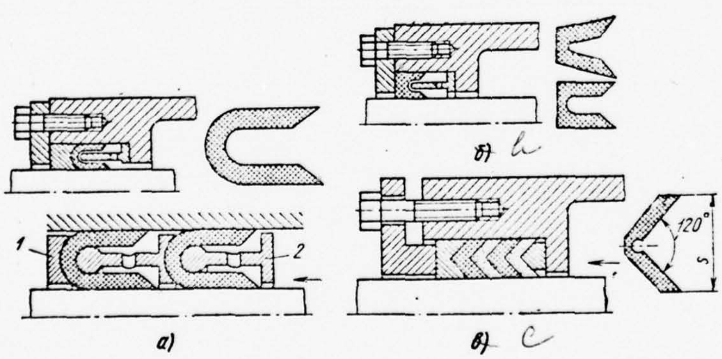

Fig. 199. Standard forms of sleeves. ¶ Furthermore, the groove must be so that of the ring, inserted in cylinder, in lock would be

preserved clearance a, necessary for the compensation for inaccuracies in the production of cylinder and distortions of its diameter on piston stroke.

Ring must free be moved in groove in axial and radial directions in order that it could be force against the conjugated/combined with it surfaces.

Cup packings. Sleeve in the general case is called the elastic figure in cross section ring (Fig. 198) from rubber compounds and their combinations with different fabrics, which, being the pressed pressure of working medium to the butting parts of the packed pair, hermetically seals joint.

The initial contact of sleeve with the packed surfaces, which ensures sealing/pressurization at zero and low pressure, is realized because of its elasticity, obtained as a result of deformation (compression) during assembly into groove (Fig. 198b). For this width a of the solution/opening of the whiskers of sleeve in free state (Fig. 198a) exceeds depth b of the groove, into which it is installed collar, as a result the whiskers collars are compressed ($b < a$),

creating close contact on ends. The density of this contact is raised with an increase in the pressure of the liquid, which expands edge collars, pressing them to packed surfaces, creating close contact (Fig. 198c).

Are most common by U-obraznye (Fig. 199a and b) and V-shaped (herringbone) collars (Fig. 199c). For packing/seal at the pressure of working medium to 350 kgf/cm² are commonly used U-shaped collars, also, in pressure to 500 kgf/cm² and above - herringbone.

U-shaped collars make with the rounded (Fig. 199a) and flat/plane (Fig. 199b) basis/base.

For the preservation/retention/maintaining of form the sleeve place during the assembly of the sealing package between shaped supporting/reference 1 and spacing 2 rings (gasket-containing) from metal or Textolite (Fig. 199a).

The number of sleeves is selected depending on operating pressure. Usually are recommended the applying of two-three collars

and only sometimes four. Page 240.

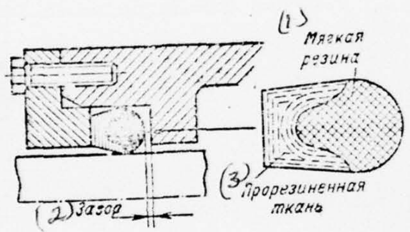

Fig. 200. Collar with the increased elasticity.

Key: (1). Soft rubber. (2). Clearance. (3). Rubberized fabric.

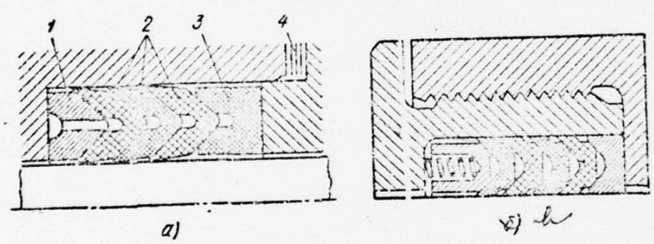

Fig. 201. Sets (packages) herringbone collar.

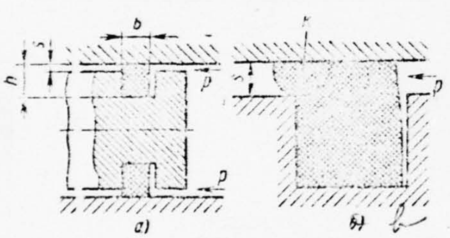

Fig. 202. Diagrams of the action of packing/seals by the rubber rings of rectangular cross section. ¶ For an improvement in the closeness of contact are applied the collars, in which the space between the solution/opening of their jaws is filled by soft rubber (Fig. 200).

Herringbone collars (Fig. 201) apply mainly for the packing/seal

of parts with reciprocating and thinner with rotary motion they make in the form of the rings, assemble/collected into package on several (3-8) pieces. The supporting/reference 3 and spacing 1 rings herringbone collar 2 (Fig. 201a) manufacture with the angles, which exceed by 5° angle of jaws collar. The control of the tightening of sleeves is realized by the appropriate selection of washers 4 or by springs (Fig. 201b). During the application/use of springs is removed the need for the manual control of the tightening of packing/seal.

Experiment shows that the packing/seal, which is of six - eight gaskets effectively prevents hydraulic slip at pressures to 400-500 kgf/cm².

Packing/seals by the rubber rings of rectangular cross section. Packing/seal of this type consists of the rectangular of cross section rubber ring, placed into the circular groove, executed of the piston body or stock/rod (Fig. 202a). The airtightness of packing/seal at the zero and low pressures of liquid is provided by the precompression of ring during assembly. For this the groove is made by such, that the ring during assembly obtains certain radial reduction, equal to 0.1-0.2 mm. During the supply of liquid under pressure p to one of the sides of ring it is displaced to side wall

of groove in the direction of effect of pressure and, transforming under the effect of this pressure, it creates close contact over three separating surfaces (Fig. 202b), whereupon the density of this contact increases virtually proportional to a pressure increase of liquid.

On the pressure of liquid depend friction and wear of ferrules. The latter is caused by the fact that during a pressure increase increases the extrusion of rubber into clearance, and also occurs the intense wear of ring as a result of trimming by its sharp edge of groove. Ring begins to be destroyed usually in the place, which borders to clearance (section k), since here besides trimming is develcped the maximum voltage of the material of ferrule with its deformation. Page 241.

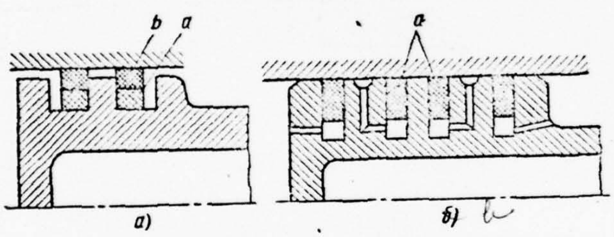

Fig. 203. Types of packing/seals by the rubber rings of rectangular cross section.

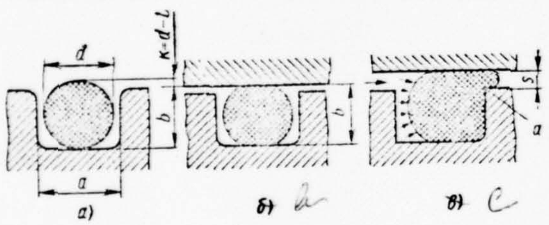

Fig. 204. Diagrams of packing/seals by the rubber ring of round cross-section.

¶ The size/dimensions of rings and grooves in piston are selected by such that during the assembly of rings in groove (with zero reduction) would be preserved the side clearance, equal to 0.2-0.25 mm. width *b* (Fig. 202a) of ring is usually equal to 3-5 mm height/altitude h was equal to 5-8 mm.

For the elimination of the possibility of the extrusion of ring into clearance they decrease the radial clearance, and also they increase the hardness of rubber. Since the latter leads to a reduction in its elasticity and to the loss of the elasticity of ring, in particular at low temperatures, and consequently, to the loss of the airtightness of packing/seal, apply two rings, arrange/located one above another (Fig. 203a). The internal (lower) ring a manufacture from the soft rubber (60-70 unity according to Shore), which retains elasticity at low temperatures, and external b from the more rubber (80-90 unity according to Shore), capable of resisting the pressure of liquid, which attempts to press ring for clearance. A similar packing/seal is useful for a work with pressure on the order of 300 kgf/cm².

For an increase in the closeness of contact of rubber ring with the packed surface into groove under ring frequently will be fed the pressure of liquid (Fig. 203b). In this case is represented possible to use rings from rubber of the high hardness (80 [90 unity according to Shore) or of teflon, thanks to which is removed the danger of the extrusion of ring into clearance. Rings a in this case are placed into grooves without side clearance, between the root and the ring is provided for the small radial clearance, able to compensate for the bloating of rubber. The compression of internal rings is realized by

pressure of hydraulic slipes through external packing/seals. Experiments on similar packing/seals show that they reliably work at pressures 350-400 kgf/cm².

Packing/seals by the rings of round cross-section. In contemporary technology are most widely common the packing/seals by the rubber rings of round cross-section (Fig. 204), the operating principle of which is analogous to the operating principle of the rings of rectangular cross section. These rings reliably and long work at pressures to 350 kgf/cm². With the preservation of ring from extrusion into clearance they are applied in pressures 1000 kgf/cm² and sometimes in pressures to 5000 kgf/cm².

The rings of round cross-section are used as in motionless, so in movable connections. For their arrangement/permutation are applied predominantly rectangular grooves. Since rubber is virtually incompressible, the volume of groove must be more the volume of ring to the value of a possible increase in the latter in operation. Page 242.

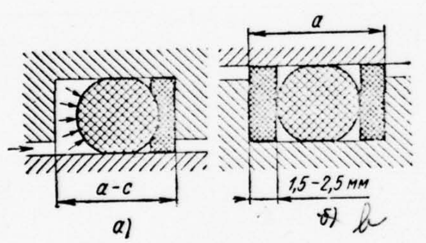

Fig. 205. Packing/seal with rubber rings with shielding adapters.

Virtually grooves on them usually are constructed with the calculation for the possible bloating of rings in working fluid within limits of 15o/o of initial volume. In the majority of cases the size/dimensions of rings and grooves in piston are selected by such that during the assembly of ring in groove (with zero reduction) would be preserved side clearance (a - d) = 0.2-0.25 mm (Fig. 204a).

For providing the required assembling compression of ring (contact stress) diameter d of its transverse section in free state and depth b of groove (Fig. 204a and b) select by such that the ring, placed into the groove between the packed surfaces, would be squeezed over cross section to value k = d - b. Packing/seal is estimated at the coefficient of the preliminary (assembling) diametric compression of the section of ring at the radial direction

$$w = \frac{d-b}{d} 100\%.$$

The reduction of rings in groove in the general case is selected equal by w = 9-13o/o.

By the indicated precompression of ring is created the airtightness of compounds at the zero and low pressure of liquid. In

the presence of pressure the ring hearth by its action, transforming at the outer side of groove, creates close contact with the packed surfaces (Fig. 204c).

Surface finish characteristics of the parts, with which contacts the ferrule of movable connection, is led for the purpose of a decrease in friction down to ∇ 9-10.

The ferrule of round cross-section transforms under the effect of pressure of liquid and under appropriate conditions, determined by the pressure of liquid, by the hardness of rubber and by the value of the packed clearance, it can be pressed into the clearance between the packed surfaces (Fig. 204c).

The indicated extrusion of ring into clearance is as for the rings of rectangular cross section, the basic reason for its destruction. With the extrusion of ring into clearance the acute angle of edge a of groove cuts into intc ring, destroying its surface (Fig. 204c).

For the elimination of the extrusion of ring into clearance the size/dimension of the latter must be as small, as this allow technological possibilities.

For the preservation of ferrules from extrusion into clearance are applied the guard rings, placed with one or along both sides of ferrule (Fig. 205a and b). Guard rings is recommended the applying of in pressures more than 100 kgf/cm².

When using guard rings the rubber packing rings of round cross-section can be applied with the pressure of order 1000 kgf/cm² and above. However, guard rings (especially leather) considerably raise (2-3 times) friction of sealing node/unit.

Guard rings can be made from any elastic material, which possesses sufficient hardness in order to resist extrusion by its pressure of liquid into clearance. Are most common rings from skin, rubber, teflon, Textolite and etc.

Calculations of rings and grooves. In the hydraulic systems of

machines in essence are applied rectangular grooves (see Fig. 204a) whose size/dimensions must be selected in such a way that during the worst combination of deviations in the size/dimensions of mating parts would be provided for the minimum assembling compression of ring. Page 243.

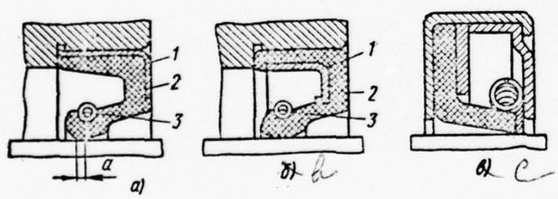

Fig. 206. Collars for the packing/seals of running shafts. The depth of the groove, into which is placed the ring, together with the clearance among the packed surfaces must be less than the diameter d poperecnogo seceni4 svobodnogo kol6qa Na velicinu k, value of which determines the value of the precompression of ring. For movable connections with the rings, which have cross-section diameter 2 mm, value k must be approximately equal to 10o/o of diameter of section

and for rings 2-6 mm in diameter it is equal to 10 - 6o/o of diameter of section. For static seals the precompression can be increased, in accordance with how value k can, if this is allow/assumed by the conditions of assembly, to be 15-20o/o of cross-section diameter of ring.

Taking into account tolerances in size of parts the actual compression of ring can be less than calculated. The actual compression of ring taking into account a change in the linear dimensions

$$k = \frac{d_{min} - b_{max}}{d_{min}},$$

where d_{min} is the minimum diameter of the section of ferrule taking into account the possible production deviations; b_{max} - the maximum depth of groove under ring.

The width of groove must be approximately to 20-25o/o greater diameter d of the cross section of ring in its free state or equal to

the width of ring in squeezed state.

PACKING/SEAL OF RUNNING SHAFTS.

The packing/seal (sealing/pressurization) of running shafts is realized by two methods: in the circumference of shaft (radial packing/seal) and over faces (end-type or mechanical packing/seal). Both methods of packing/seals are constructed on contact principle.

Radial type packing/seals. In machine-building won acceptance of radial (sleeve) type packing/seal (Fig. 206). For production a collar utilize rubber, resin-like materials and thinner - skin.

Figure 206a and b depicts the design concepts standard collar from rubber and Fig. 206c - from skin. Packing/seals with rubber gaskets (Fig. 206a and c) differ from each other in terms of the place of the location of the metal frame (ring of hardness) 1, of employee for an increase in the hardness collar 2. The framework/body is arrange/located from outer (Fig. 206a) and internal (Fig. 206b) side collar, and also it is installed inside sleeve. Framework/body

usually is connected with sleeve by vulcanization.

The special feature/peculiarity of the work of the packing/seals of running shafts is the fact that the contact sealing collars with the surface of the packed shaft occurs over small surface, in consequence of which on this surface and the contacting with it sealing edge the collars are developed high temperatures. In view of the fact that with a pressure increase of the packed medium contact pressure and friction increase, the sealing collars in question are applied in the pressures of the liquid before the packing/seal not above 1-2 kgf/cm². Page 244.

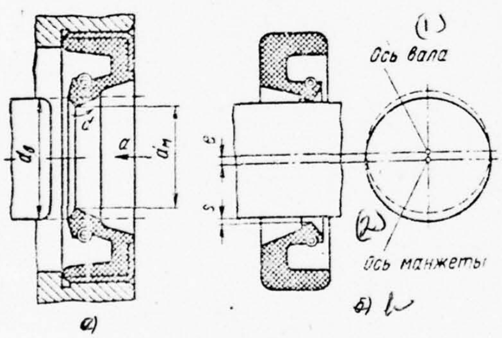

Fig. 207. Schematic of setting sleeve on shaft.

Key: (1). Axle/axis of shaft. (2). Axle/axis of sleeve. ¶ Collar it must be establish/installed to shaft with tension, which is reached by the fact that the diameter d_M opening/aperture in collar

in free state is selected less than the diameter d_B shaft (Fig. 207a). In practice the size/dimensions collars select so that during its assembly to shaft the inner diameter of the sealing jaws would be increased by 5-8o/o.

For providing a reliable contact of ring with shaft apply the supplementary pressing the collars, which is realized with the aid of spiral (band) spring 3 (see Fig. 206a and b). The inner diameter of the ring of unloaded spring usually approximately to 2 mm is less than the working diameter of groove on collar under spring, equal to diameter d_B shaft plus the doubled thickness collars. During calculations they proceed in order that with the minimum diameter shaft, the maximum thickness $d_{B\,min}$ collars and the maximum inner s_{max} diameter $d_{0\,max}$ the ring of spring would be provided for the elongation of spring not less than on 1 mm at the diameter:

$$d_{B\,min} + 2s_{max} - d_{0\,max} \geqslant 1.$$

With radial play of the packed shaft the airtightness of packing/seal unavoidably is disrupted. This is caused by the fact

that for the preservation/retention/maintaining of the closeness of contact of sleeve with shaft it is necessary to ensure the continuous coupling of edge collars with the surface of shaft during its rotation/revolution. From the diagram, given Fig. 207b it is apparent that shows with the eccentric location of the rotational axis of the shaft of its relatively geometric axle/axis the shaft accomplishes gyration with the amplitude, equal to eccentricity e. In this case the points of contact of the edge of sleeve with shaft accomplish as a result of the eccentricity of the rotational axis of shaft motion along oval (elliptical) trajectory. If the edge of sleeve does not manage as a result of the action of the forces of inertia and friction, or insufficient elasticity of sealing cell/element to follow (" to follow") after the surface of shaft, then between it and the shaft is formed clearance s, instantaneous position of which will be changed in each revolution of shaft by 360°. The possibility of the formation/education of this clearance and its size/dimension are determined by eccentricity e and especially by the frequency of the rotation/revolution of shaft.

Packing/seals by the rings of round cross-section. For the packing/seal of running shafts can be applied also the rings of round cross-section; however, during the setting up of these rings according to usual diagram (square with the rotational axis of shaft)

they can be applied only in small peripheral speed (to 2.5 m/s) and the radial compression of ring w not more than 5-6o/o.

These limitations are caused by the fact that on the contact surface of these rings are developed inadmissibly the high temperatures, calling the rapid output/yield of packing/seal of system.

To lower friction and to facilitate work conditions is possible with the setting up of rings at certain angle (Fig. 208) for the plane, perpendicular to the axle/axis of shaft, thanks to which considerably they are improved the lubricant of the friction surfaces and condition of the diversion/tap from them of heat. The lubricant in this case enters the contact zone forcedly and with each revolution of shaft is renewed. Furthermore, with the inclined location of ring the zone of friction is not limited by the ridge of contact, but seemingly had spread by ring in its relative motion, as a result ring covers wider the section of the surface of running shaft, thanks to which considerably it is improved the diversion/tap of heat from rubbing surface. Page 245.

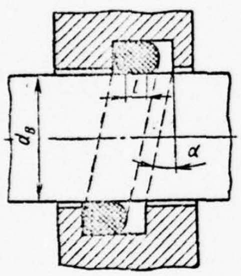

Fig. 208. Sealing/pressurization of running shafts by the rubber ring cf round cross-section.

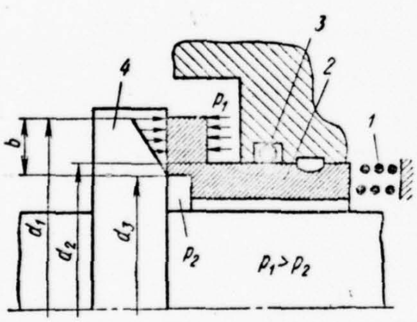

Fig. 209.

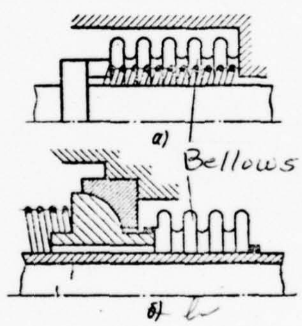

Fig. 210.

Fig. 209. Diagram of end-type packing/seal.

Fig. 210. Diagrams of end-type packing/seals with bellows.

End-type type packing/seals. In connection with an increase in the requirements for the packing/seals of running shafts (work with high pressures and the revolutions of shaft at high temperatures) arose the need for the investigation of the new diagrams, corresponding to these conditions.

Most completely these requirements answer end-type type packing/seals (Fig. 209), in which the driving packing surface contacts with the external surface of shaft in the plane, perpendicular to the axle/axis of shaft. These packing/seals provide high, virtually complete airtightness and large service life, and also they differ in terms of relatively low power losses to friction. Packing/seals can be applied in the peripheral speed of the packed node/unit to 60 m/s and the pressures of the packed medium to 400 kgf/cm².

End-type packing/seal (Fig. 209) consists of spring 1, ferrule 2, prepared from soft antifriction material, and which contacts with it on the end/face of the metallic carrier ring (bush) of 4 high hardnesses. Ferrule hermetically sealed connected either with running shaft or with stationary casing. Carrier ring in the first case is placed in housing and in the second - on running shaft. In this case

one of the rings must free be mixed along axle/axis. Therefore it by spring 1 can be pressed to the second ring. With the aid of spring is created preliminary contact face pressure of rings, sufficient for the prevention of hydraulic slipes at zero or the close to it pressures of working medium. With an increase in the pressure to the effort/force of spring 1 is added the effort/force of the unbalanced pressure of liquid in the chamber from spring, thanks to which contact pressure (specific load) that which slide the vapors it will be raised proportional to an increase in this pressure.

The packing/seal of the radial slot between movable cell/element (ring) 2 and housing is realized by a circular rubber ring by 3 (Fig. 209) or other ferrules and by collars, and also bellows (Fig. 210a and b).

The reliability of the operation of sealing equipment/devices in question first of all depends under otherwise equal conditions on the correct relationship of the area of the contact surface of rings F and of area f, on which acts the pressure of liquid, which loads the moving cell/element (ring) of packing/seal, and also partially from the effort/force of spring, initially loading this cell/element.

Page 246.

For a reduction in the contact pressure the area

$$f=\frac{\pi}{4}(d_1^2-d_2^2),$$

on which acts the pressure of the liquid, which presses slider to motionless, is selected less than the area

$$F=\frac{\pi}{4}(d_1^2-d_3^2),$$

on which it occurs contact the vapors (see Fig. 209).

By the correct selection of the relation

$$k=\frac{f}{F},$$

which is called the coefficient of the balancing of packing/seal, it is possible to obtain the contact pressure of rings considerably lower than the pressure of working medium.

Assuming that the direct contact of the surfaces of the sliding pair is absent (acting forces are balanced) and flow of liquid in clearance it is subordinated to hydrodynamic law, equilibrium condition of the axial forces of those applied to movable under axial direction cell/element 2, will take the form

$$p_{cp}F=\Delta pf\pm T-P_{np}=0, \qquad (69)$$

where p_{cp} is the mean pressure of liquid in the clearance between the adjacent surfaces of rings; $F = \pi\ (d^2{}_1 - d^2{}_3)\ /4$ - the area of the adjacent surfaces; $\Delta p = p_1 - p_2$ - the pressure differential between the packed medium and low-pressure cavity (when low-pressure cavity is connected with the atmosphere, $\Delta p = p_1$); $f = \pi\ (d^2{}_1 - d^2{}_2)$

/4 - the area, on which acts the pressure of liquid, which presses moving cell/element the vapors to motionless; T acts the frictional force of movable cell/element; P_{np} is an effort/force of the tightening of spring.

In view of the fact that of the correctly designed packing/seal sum T + P_{np} usually does not exceed 5-8o/o of axial force of the pressure of liquid, which acts on moving cell/element, in the calculations by it usually they disregard. Under this assumption equation (69) will take the form

$$p_{cp}=p_1\frac{f}{F}=\frac{p_1\left(d_1^2-d_2^2\right)}{d_1^2-d_3^2},$$

or

$$\frac{p_{cp}}{p_1}=\frac{d_1^2-d_2^2}{d_1^2-d_3^2}.$$

Page 247.

Allow/assuming further, that pressure distribution of liquid in clearance in radial direction in the width

$$b = \frac{d_1 - d_3}{2}$$

packing surface (band) will be linear, which will be correct under

the condition of the parallelism of the surfaces, which form clearance, it is possible to accept

$$p_{cp} \approx \frac{p_1}{2}.$$

Accordingly, the equilibrium of forces of the pressure of liquid on ferrule taking into account the indicated wedging action of liquid during the linear distribution of clearance pressure will set in under the condition

$$k = \frac{f}{F} = \frac{d_1^2 - d_2^2}{d_1^2 - d_3^2} = 0{,}5,$$

where k - the coefficient of balancing (discharging).

At this coefficient of k of balancing the closeness of contact, required for the preservation/retention/maintaining of airtightness, is achieved only by the action of the stress force of spring 1.

Since the distribution of clearance pressure according to a radius can be nonlinear, the coefficient of balancing usually selects $k = 0.6$, since otherwise the effort/force of the pressure of liquid in clearance can exceed the effort/force of pressing rings and packing/seal "will be opened".

During the taken condition $k > 0.5$ will appear the surplus of power, which presses slider to motionless.

The airtightness of end-type packing/seal greater than other packing/seals, depends on the accuracy/precision of production and quality of sliding surfaces. The most important value, and in particular at the high slip rates, has an observance of the perpendicularity of the sealing plane to the rotational axis of shaft.

The permissible end-type play depends on velocity, which is caused by the fact that if with small revolutions the movable in axial direction ring can in full or in part compensate some disturbances of the perpendicularity of contact surfaces to rotational axis, then at the high frequencies of rotation/revolution per minute this compensation due to the action of the forces will become impossible, and ring as a result of the formed tapered clearance had lost closeness of contact, i.e., with certain end-type play the orienting ring as "jumps", retaining contact with carrier ring not over an entire surface, but only at one point.

Great effect on the airtightness of packing/seal exerts the flatness of the contacting (workers) surfaces of tracks, deviation from which both during the production and in operation must not exceed 1 - 0.5 μm by a radius 50 mm. Most rational is the purity/finish of treatment/working the working surfaces of ferrules on requirements ∇ 10.

The given requirements for the minimum of end-type play and parallelism of working surfaces partially can be lowered during the application/use of packing/seals with spherical (see Fig. 210b) rings.

For the production of the parts of end-type packing/seal are utilized the materials, used in bearings and step bearings of slip. Specifically, is common the pair from bronze and cast iron ferrules and steel carrier ring (bush) with the cemented surface.

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[redacted]

[redacted]

[redacted]

Page 248.

Chapter VII.

Decontamination (filtration) of working fluid.

The contamination of liquids by different impurity/admixtures lowers reliability and the service life of hydrcaggregates, whereupon

the effect of the quality of the decontamination (filtration) of liquids on the work of hydroaggregates is sc/such great, that without exaggeration it is possible to confirm that the service life of hydraulic machines and hydroaggregates can be increased or lowered/reduced depending on the quality of the decontamination of liquids several (to ten) once. The particles of contaminator, as a rule, raise friction and can lead to the wedging of the movable parts of hydroaggregates, and also be the reason for the galloping motion of exit component/link during a smooth change in the control signal. Solid, and in particular the abrasive particles, caught into clearance, cause the wear of working surfaces sliding the vapors during its motion.

Filter is called equipment/device, in which the liquid undergoes decontamination from the solid and viscous contaminating impurity/admixtures, which fall into hydraulic system from without (specifically, with airborne dust), and also the forming as a result of wear parts of hydroaggregates and aging of liquids. From solid particles destructive for hydroaggregates are the particles, which form part of airborne dust, which fall into the tank through different channels.

Obvious, to achieve the absolute purity/finish of liquids with the existing methods of their decontamination is impossible. In view of this during the solution to the question concerning requirements for the quality (fineness) of filtration it is necessary to be guided by experimental data and recommendations. Virtually filtration is considered satisfactory, if the size/dimension of the capillary channels of the filtering material does not exceed minimum clearance by the sliding vapors of the hydroaggregate, for which is intended the filter. These requirements in the general case they satisfy filters with the fineness of decontamination 5 μm and for the critical hydraulic systems (control systems with servo valves, etc.) - with the fineness of decontamination 3 μm.

FINENESS OF FILTRATION.

The material of filter cells must be maximally penetrated; however, capable of detaining the smallest possible particles of mechanical connection/inclusions. Accordingly, it must have the smallest uniform grid with maximum as the area of passage cells (pores) and by their amount per the unit surface area of material. The amount of these cells per unit of surface determines the fineness of filtration, which is characterized by the size/dimensions of pore

channels in filter material or, which is the same, by the sizes of particles of the contaminator, which are held by cleaner.

Page 249.

Therefore by the fineness of filtration is understood the minimal size of particles of the contaminator of the filtered medium, recovered by filter cell, i.e., the ability of filter to detain (to drive out) from the liquid of the particle of the specific size/dimensions.

In accordance with requirements for the fineness of the decontamination of liquids divide the filters of coarse, normal, fine/thin and especially fine purification, the restraining particles of contaminator with a conditional diameter of more than 100; 10; 5 and 1 μm.

In industrial markings and under technical specifications for filters usually they indicate the minimum (nominal) size of particles, which are detained by this filter. Thus, for instance, "10

- micrometric" the filter is defined as filter, which must provide removal/distance in one operation/pass 98o/o (throughout mass) of all particles with size/dimension on the greatest measurement, equal or large 10 μm. The better/best specimen/samples of series filters provide the fineness of filtration 5 μm.

METHODS OF FILTRATION AND TYPES OF FILTERS.

Isolation/evolution from the liquids of the solid contaminating impurity/admixtures is realized by mechanical or power methods. In the first case the filtration is realized applying different slotted and porous filter cells (materials), and in the second - by the application/use of force fields - magnetic, electrical, centrifugal, etc.

In the hydraulic systems of machines is applied predominantly the first purification method, with which from liquid with passage by it through the filter cell are separate/liberated the particles as a result of a difference in the size/dimensions of these particles and passage capillary channels of filter material.

Metallic wirings. When to filters are not presented high requirements with respect to the fineness of decontamination, are applied the metallic woven meshes of square entanglement from the wire (by predominantly brass) of round cross-section.

The filtering qualities of these filters (fineness of filtration and fluid flow rate) are characterized the size of cell in world/light and "density" or by the area of the living (passage) section of cells into units of surface area. The last/latter parameter expresses as the coefficient of clear opening b, which is the ratio of the area of the passage cells F_0 to the common/general/total area F of the mesh:

$$b=\frac{F_0}{F}=\left(\frac{a}{a+d}\right)^2,$$

where d is a diameter of the wire of mesh; a - the size/dimensions of the side of mesh in world/light.

Decrease under all other identical conditions of the value of cell is accompanied by a decrease in the coefficient of the clear opening of mesh, and, as a result, by an increase in the hydraulic friction of filter. Filter elements from meshes are fulfilled in the form of cylinders with the corrugated or smooth surface (Figs. 211 and 212), and also in the form of the set (package) of mesh disks (Fig. 213) and etc. In order to avoid the failure of mesh in the case of its blockage by the filtered out inclusions, in the casing of filter is placed bypass valve a, which during the blockage of filter element and an increase in this case in the pressure differential on it is open/disclosed, and liquid proceeds to exit branch, passing filter element.

Gauze filters frequently are fulfilled also with several (by two and three) layers of filtering meshes with the constant in all meshes size/dimensions of cells or the meshes, the size/dimension of cells of which decreases from one layer to the next in fluid flow (Fig. 212). The application/use of filters with multilayer meshes considerably raises effectiveness and the fineness of decontamination. Page 250.

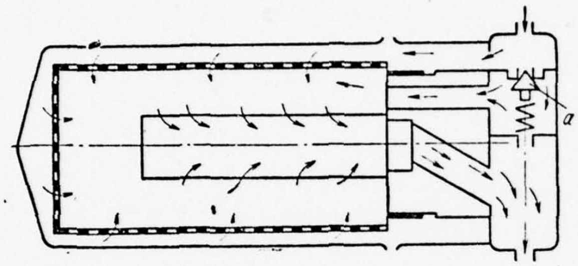

Fig. 211.

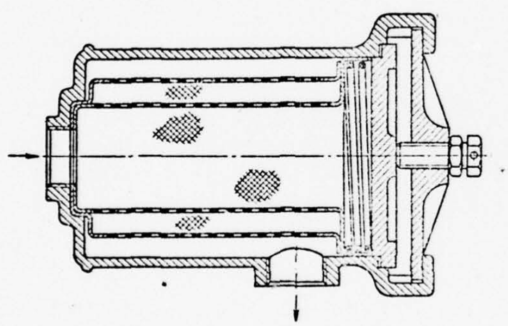

Fig. 212.

Fig. 211. Diagram of gauze filter.

Fig. 212. Diagram of two-layered gauze filter.

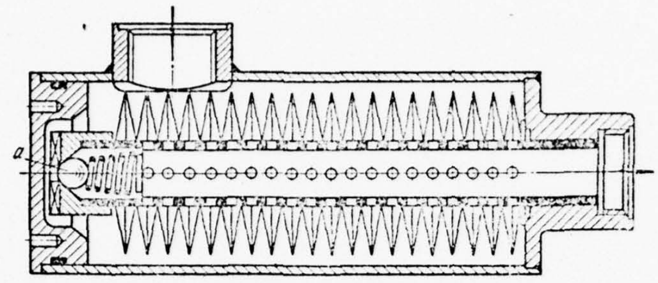

Fig. 213.

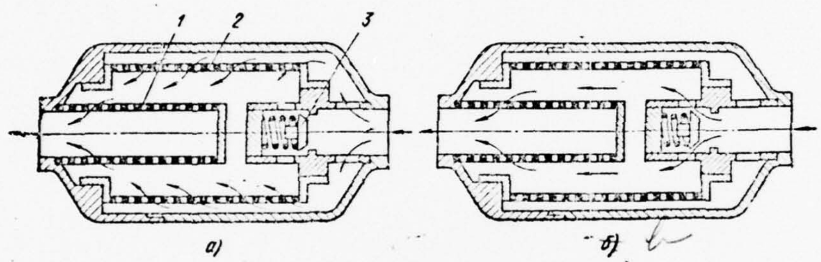

Fig. 214.

Fig. 213. Filter from mesh disks.

Fig. 214. Rejector-acceptor circuit from the cell/elements of coarse and fine purification. Page 251.

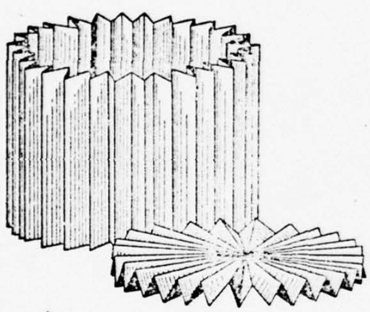

Fig. 215.

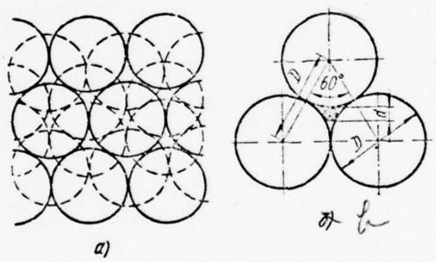

Fig. 216.

Fig. 215. Paper filter elements.

Fig. 216. Structure of filter material from the sintered ball/spheres (a) and the design diagram of filter element (b). ¶ The fineness

of filtration by these filters depends on the size/dimension of mesh in the world/light, minimum value of which for the meshes of simple entanglement is equal to to 0.08-0.1 mm.

In the hydraulic systems of some machines (airplanes, etc.) apply the nickel filtering meshes of the complex braiding (twill, etc.), better/best specimen/samples of which filter out particles 2-3 μm in size/dimension. These meshes consist of several (5-10) layers of the twisted wire, between which are laid the cell/elements from woven wire. The wire diameter is frequently equal to several micrometers.

Filters with paper cell/elements. Filters with paper and cloth cell/elements detain in one operation/pass the considerable (75o/o) part of hard spots whose size/dimension is more than 4-5 μm. Filters they supply with bypass valve (Fig. 214).

In order to exclude the possibility of the incidence/impingement into system in the case of discovery/opening the bypass valve of unfiltered oil, filters they supply with the supplementary cell/element of rough cleaning. The diagram of this filter with the

combined cell/element, which consists of the cell/elements of fine/thin 2 and coarse 1 decontamination, is represented in Fig. 214a and b. To discovery/opening bypass valve (Fig. 214a) liquid consecutively it passes through both cell/elements. During the blockage of the cell/element (Fig. 214b) of fine purification is open/disclosed bypass valve 3, and the liquid through the cell/element of rough cleaning proceeds to exit branch, passing the cell/element of fine purification.

Paper cell/element usually is made in the form of the cylinder whose walls for an increase in the filter surface area are assemble/collected into the folds of one form or the other, supported with metal frame (Fig. 215).

Deep filters. The filters, in which liquid it passes through the thickness of porous material (filler), are called deep. The filters of this type, each capillary of which has the large amount of consecutive pores, which reaches to hundred and more, it is possible to compare on the effectiveness of filtration with the multilayer filters of surface types with the same length of capillaries and the amount of pores in them. Since contaminator is detained in these filters in essence in the pores of the thickness of material, these

filters with the identical contamination of liquid have in comparison with surface filters higher dirt capacitty and service lives.

Are widely common deep type filters with the fillers from porous metals and ceramics, obtained by sintering metallic spherical and nonspherical powders.

The diagram of the porous structure of cermet filtering material is represented in Fig. 216a. Page 252.

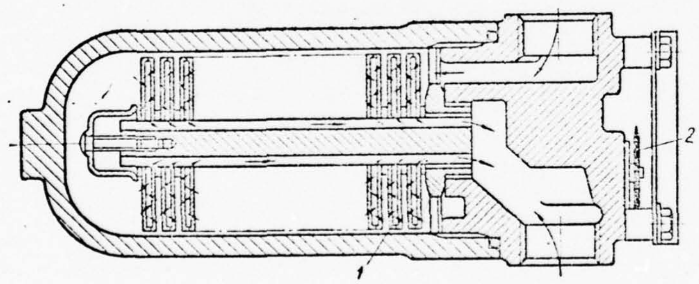

Fig. 217. Filter from cermet disks. [continues] Liquid is cleaned, occur/flow/lasting over the long and winding channels between ball/spheres, whereupon the delayed contaminator is distributed virtually evenly on these channels and pores, thanks to which filters they differ in terms of high dirt capacity.

The size/dimensions of the pores of cell/elements from powdered metal and ceramics are selected on the basis of the maximum conditional diameter of the particle of contaminator, which can pass in the clearance between three by tightly packed spheres with the spherical form of the initial powder and the point contact of ball/spheres the maximum linear size of pores (maximum conditional diameter of the particle of contaminator, which can traverse the pore) it is possible to calculate according to expression (Fig. 216b)

$$d = 0{,}155D,$$

where D - the diameter of grain (ball/sphere) of the initial powder.

In actuality, the linear dimension of pores as a result of the roughness of grains, deviation from their correct spherical form than less indicated. Taking this into account the actual size of the pores

$$d \approx 0{,}1D.$$

Is represented possible to obtain the minimal size (diameter D) of the sphere of metallic initial material approximately 5 μm, which corresponds to the conditional diameter d of the pores of cell/element 0.5 μm.

This conditional diameter of pore determines the nominal fineness of the decontamination of liquid. However, experiment shows that these filters detain the considerable amount of particles whose size/dimensions are less than the nominal (conditional) size/dimension of pores. The latter is caused by the fact that the particles with a size/dimension of the undersize of pores are detained in the narrowing themselves labyrinths of the space lattice (porous structure) of material.

The advantages of metallic filtering material include also the

fact that they allow/assume machining, pressing, sintering, and majority - soldering, thanks to which of them it is possible to manufacture the cell/elements of any requiring form.

In the majority of cases these cell/elements are manufactured in the form of the corresponding form of briquettes or sheets 0.4-1.0 mm thickness, from which can be made the cell/element, analogous in construction paper.

Figure 217 shows filter with the element of lamellar structure. Disks 1 on periphery are welded by argon-arc welding. In each disk there is to 100 layers of ball/spheres. Filter is equipped with bypass valve and the indicator of 2 contaminations in the form of the red knob/button whose lift testifies to the contamination of filter more than 50o/o.

Calculation of filters. The calculation of filter is reduced to the determination of flow of liquid and hydraulic friction (losses) per the unit surface area of filtering material.

Page 253.

In view of the complexity of the porous structure of the majority of filtering materials, which consists of the connected between themselves pores and the complex grid/network of the channels (capillaries) of various forms and size/dimensions, which moreover in a number of cases change under the action of pressure differential, to establish/install for the majority of filtering materials regularity and to give analytical expression for the stream conditions of liquid is virtually impossible. In view of this the hydraulic characteristics of the filtering material are determined, with the exception of individual cases, experimentally.

The specific capacity q $l/(\mathrm{min}\cdot\mathrm{cm}^2)$ and respectively the expenditure/consumption of liquid in filter Q l/min can be expressed by the dependences, which escape/ensue from poiseuille equation:

$$q = k\frac{\Delta p}{\mu};$$

$$Q = qF = k\frac{\Delta pF}{\mu},$$

where $k = q\mu/\Delta p$ is expressed the proportionality factor, which is the specific capacity of the unit surface area of filtering material in 1/ (min•cm²) with a jump/drop in the pressure 1 kgf/cm² and in the dynamic viscosity of liquid 1 poise; μ - the dynamic viscosity of liquid being filtered in poise; Δp - the pressure differential on filter in kgf/cm²; F is a surface area of cell/element in cm².

Experiment shows that coefficient k for this filter material is retained under otherwise equal conditions in practice constant over a wide range of expenditure/consumptions and the pressure differentials, thanks to which is represented possible to utilize it as a comparative evaluation of the hydraulic friction of this material. The value of this coefficient is given in the plant

characteristics of materials.

Diagrams of filtration and the site of installation of filter. They filter either entire fluid flow or part of it. The first diagram is called the diagram consecutive, and the second - by the diagram of the parallel connection of filter.

The sequential switching circuit of filter provides the filtration of an entire liquid, which participates in circulation. filter in this case must be designed to complete fluid flow rate.

The filtration of part of the flow is commonly used when are presented the requirements for especially careful decontamination of the liquid, which enters the critical hydroaggregates, and also for the preventive fine purification of the liquid of hydraulic system. For the filtration of part of the flow are commonly used deep fine filters.

In the majority of cases it is expedient to apply simultaneously both diagrams of the filtration: for the filtration of entire flow to

apply the filter, which has relatively high porosity, and for the protection of the especially critical assemblies - fine filters.

When selecting place for the installation of the filter of complete expenditure/consumption, they are guided by the following considerations. For the preservation of the pump, which is most sensitive to the contaminations of liquid filter desirable to establish/install in the suction line of pump (Fig. 218a). However, in view of the fact that filter increases the friction of the suction line and makes the conditions of the filling of pump with liquid, this mounting method of filter worse under hydraulic systems with self-priming pump is not common.

During the installation of filters on delivery line (Fig. 218b) there can be obtained higher friction. The casing of filter in this case will be located under operating pressure.

Is applied also the installation of filter in the drain line (Fig. 218c), in which through the filter it passes liquid also in the periods of discharging pump. Page 254.

Page 1018

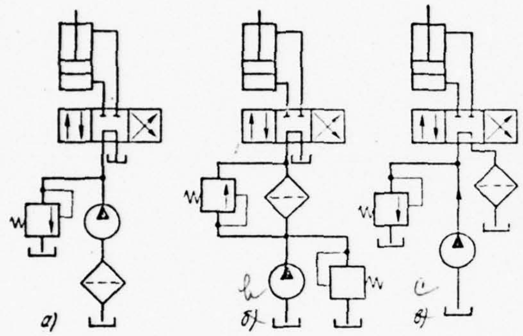

Fig. 218.

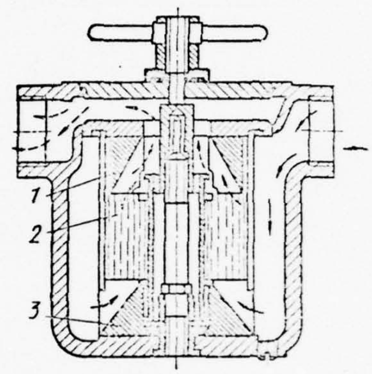

Fig. 219.

Fig. 218. Diagrams of the arrangement/permutation of filter in hydraulic system.

Fig. 219. Knife-edge filter with magnetic insets.

Magnetic filters of liquid. For trapping ferromagnetic particles are applied also the magnetic filters, which usually combine with any slotted (porous) filter. First stage of such rejector-acceptor circuits is the magnetic cell/element, which delays (recovering) ferromagnetic particles, and the second is the porous filter, which detains the diamagnetic contaminating particles, and also the ferromagnetic particles, which were being detached away from first (magnetic) stage. The application/use of a magnetic field in the similar rejector-acceptor circuit raises also the fineness of the filtration of porous filter.

Magnetic field usual is created by several (two-three) permanent magnets, fastened from the outer side of the casing of the filter, which in this case is manufactured from material with high magnetic permeability.

Figure 219 shows the rejector-acceptor circuit, which consists of mesh filter package 2 and two established/installed on entrance

and exit permanent magnets 1 and 3, which recover ferromagnetic particles. Liquid it passes through the groove/slots of lower magnet inside cylinder with mesh package and emerges through the groove/slots of the upper magnet.

Magnetic filters detain the smallest ferromagnetic particles (0.4 μm and less), which cannot be separate/liberated by mechanical filters. Simultaneously with this magnetic filters recover also the nonmetallic particles of contaminator, which being connected in magnetic field together with ferromagnetic particles form the easily separate/liberated large particles.

The centrifugal filters of liquid. In the hydraulic systems of a series of machines are applied the centrifugal filters of liquid (centrifuges), which clean liquid from the contaminating particles with the density, which exceeds the density of liquid.

The schematic diagram of centrifugal filter is represented in Fig. 220a. The liquid, which is subject to decontamination, is supplied through the tubular axle under pressure 3-6 kgf/cm^2 in rotating rotor 1, in which it untwists to certain speed, close to the

speed of rotor. In this case mud impurity/admixtures (particle) with density exceeding the density of oil, are reject/thrown by the action of centrifugal force to the walls of rotor and are precipitated out on them.

The centrifugal force, which acts on the particle of contaminator, which rotates together with by that filled by liquid by rotor,

$$F_{ц} = (m - m_{ж}) r\omega^2 = V(\rho - \rho_{ж}) r\omega^2,$$

where $m = V\rho$ and $m_{ж} = V\rho_{ж}$ - the mass of the particle of contaminator and liquid; ρ and $\rho_{ж}$ - the particle density of contaminator and liquid; V - the volume of the particle of contaminator; r is the current radius, i.e., the instantaneous distance of the center of gravity of particle of the spin axis of rotor; ω is the angular rate of rotation of particle around the axle/axis of rotor. Page 255.

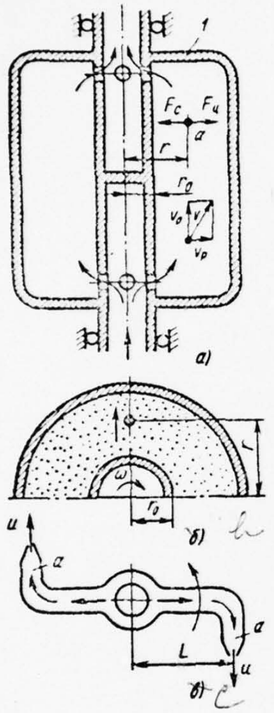

Fig. 220. Diagram of centrifugal filter.

Accepting, that the angular velocities of liquid and rotor are equal, can write

$$\omega = \frac{\pi n}{30},$$

where n is frequency of the rotation/revolution of rotor in r/min.

Under the action of centrifugal force $F_{ц}$ the particle of contaminator, overcoming resistance of liquid F_{c}, is moved in radial direction at certain rate of deposition v_{p} (Fig. 220a). Simultaneously particle is moved together with the liquid through the rotor in axial direction with the rate

$$v_0 = \frac{Q}{S},$$

where Q - fluid flow rate through the rotor; S is a flow passage cross-sectional area of rotor (area of its clearance space).

In summation, the particle is moved at resultant velocity v at certain angle to the axle/axis of rotor, approaching its wall. Upon reaching of wall it deposits on it. During this particle motion experience/tests the hydraulic friction, radial component F_c of which can be calculated for a spherical particle according to Stokes' formula (inertia strengths of particle we disregard)

$$F_c = 3\pi\mu v_p d,$$

where μ - the dynamic viscosity of liquid; d is a conditional diameter of particle.

For the case of uniform particle motion is correct the equality

$$F_ц = F_c.$$

accordingly

$$V(\rho - \rho_{ж})\,\omega^2 r = 3\pi\mu v_p d. \qquad (70)$$

Under condition $F_{u} < F_{c}$ the particle is not precipitated out on the wall of centrifuge.

The minimum diameter d of the spherical particle of contaminator, which is precipitated out at the given speed in the rotor of cleaner in one operation/pass through it of liquid, we find from expression (70):

$$d = \frac{V\omega^2 r}{3\pi\mu v_p}(\rho - \rho_{ж}).$$

Practical interest, and in particular at the high velocities of the rotation/revolution of centrifuge, can present the calculation of the pressure of liquid on its walls. When the liquid rotates at the same angular velocity, as centrifuge, each particle, which is located at a distance r from center of rotation, is subjected to the action

of the centripetal acceleration:

$$a = r\omega^2.$$

Page 256.

Pressure p, developed on radius r, is calculated from the expression

$$p = \frac{(\rho - \rho_{ж})\, r^2\omega^2}{2} = \frac{(\rho - \rho_{ж})\, u^2}{2},$$

where u = ωr - the peripheral (linear) velocity of liquid on radius r.

In centrifuge the liquid fills not cylinder, but the annular space ~~of the mass of liquid (it is noted by points)~~ (Fig. 220, B). In this case the pressure at any point in the rotating annular mass of liquid (is indicated by points).

$$p = \frac{(\rho - \rho_{ж})(r^2 - r_0^2)\,\omega^2}{2}.$$

According to the type of the drive of centrifuge (rotors) it is possible to divide into filters with hydrojet and with power drive, whereupon are most common centrifuges with the hydrojet drive, constructed according to the principle of Segner's wheel (Fig. 220c). The purified liquid from the rotor through the hollow exit axle/axis of rotor proceeds to two arrange/located tangentially the axle/axis of rotor and diametrically opposite each other to nozzles (nozzles) a. The reaction forces of fluid flow, which escape/ensues from these sopel, create the torque/moment, which gives rotor with its filling liquid in rotation/revolution with the frequency, which can be led to 6000-7000 r/min.

The reaction force of fluid flow, which escape/ensues of one

nozzle, according to expression (17)

$$R = \frac{m}{2}(u - v_{con}) = \frac{Q}{2}\rho(u - v_{con}),$$

where m = $Q\rho$ - the mass fluid flow rate per second for drive sopel (escape/ensuing from both nozzles); Q is fluid flow rate for the drive of centrifuge (expenditure/consumption through both nozzles per second); ρ - the density of liquid; u - the jet velocity of fluid flow at the nozzle outlet; V_{con} is the peripheral speed of nozzle.

Taking into account equality (20), we will obtain

$$v_{con} = \omega L = \frac{\pi n}{30} L$$

and can write

$$R = -\frac{Q}{2}\rho\left(\sqrt{\frac{2\,\Delta p}{\rho}} - \frac{\pi n}{30}L\right) = \frac{Q}{2}\rho\left(\frac{Q}{2\mu f} - \frac{\pi n}{30}L\right),$$

where μ is a coefficient of the expenditure/consumption of nozzle (it is possible to accept μ = 0.9); f - the sectional area of exit nozzle hole; n - the frequency of the rotation/revolution of rotor (centrifuge) in r/min; L is a distance from the axle/axis of OT nozzle to the spin axis of rotor.

The torsional moment, developed with the hydrojet drive, which consists of two sopel,

$$M_p = 2RL = m(u - v_{con})L = Q\rho L\left(\frac{Q}{2\mu f} - \frac{\pi n}{30}L\right).$$

Since virtually it is not possible to obtain the large values of reaction force, filters with hydrojet drive they cannot ensure the high angular velocities of rotor (frequency of the rotation/revolution of rotor with hydrojet drive is limited 6000-7000 r/min), and consequently, they cannot ensure the high fineness of the decontamination of liquid, which is virtually equal to 20-30 μm. In view of this are applied the centrifuges with mechanical and electric drive, the velocities of rotor of which are led in some constructions, if this is admissible according to the conditions of the acceleration of liquid under rotor, to 20,000 r/min.

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Pages 257-280.

Chapter VIII.

Pneumatic (gas) drives.

In contemporary machines, and in particular in the systems of the automation of production processes, along with hydraulic

mechanisms are applied the pneumos-mechanism (pneumatic actuators), based on the use as working medium of the compressed or rarefied air (in the present course are examined only the first types of mechanisms).

With the aid of pressure-operated devices (drives) are solved the complex problems in the automation of the administration of machines and production processes. Their application/use has advantages when the application/use of hydraulic drives with oil working medium is inadmissible on the requirements for fire safety, as it takes place in carbon shaft/mines and in a series of chemical production.

The major advantages of pressure-operated devices include the reliability and life, the speed of action (function), simplicity and cost-effectiveness/efficiency, caused by the single-channel power supply of actuating pneumos-mechanism (mastered air is discharged directly in the atmosphere without drain pipes) and by the cheapness of quite working medium.

Along with the positive qualities of pneumatic system they

possess the number of the deficiency/lacks, which escape/ensue from the nature of working medium - air. Air possesses high compressibility, in view of which it during compression accumulates the energy, which under known conditions can be converted into the kinetic energy of the driving masses and cause impact loads.

Because of this the pneumatic power-distribution systems do not provide without the special supplementary means for the necessary evenness and accuracy/precision of course. The compressibility of air in pneumatic systems excludes the possibility of direct fixation of controls in the assigned intermediate positions. In equal measure in pneumatic actuator is difficult obtaining with the varying load of uniform and stable velocity.

Besides this pneumatic actuators they have, as a rule, lower efficiency in comparison with hydraulic drive, and also require the application/use of lubricators.

The compressed air for the power supply of pneumatic systems usually is developed by the compressors, attendant of entire enterprise or their determined group. In the centralized and group

power-supply systems usually is applied the pressure 5-6 kgf/cm², in individual power supply - to 50 kgf/cm² and above.

Working medium/propellant in pneumatic actuators is compressed air; therefore the calculation of processes in this drive is based on the laws and equations gas- and thermodynamics.

Page 258.

Since the questions gas- and the thermodynamics, placed as the basis of gas-dynamic calculations of pneumatic systems and their cell/elements, are examined in the previous training courses "gas hydrodynamics" and "thermodynamics", in the present course are examined the circuits of action, construction of pneumatic actuators and their cell/elements, and also the methods of the engineering calculations of these cell/elements. The questions of gas dynamics are given in the form of reference data in the volume, necessary for mastering the material of the present course.

PARAMETERS OF STATE OF THE GAS.

The processes of compression and expansion of air during its flow in the channels of pneumatic systems are accompanied by changes in the parameters of its states, basic from which are the pressure p, the temperature T and specific volume v. Pressure enters, with the exception of the cases, specified especially, all the given thermo- and gas-dynamic dependences in absolute unit.

Specific volume v (volume, occupied by the unit of the mass of gas) is connected with volume of V gas by the dependence

$$v=\frac{V}{m},$$

where m - the mass of the gas, included in volume of V.

Since V = m/ρ, we can write

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$$v=\frac{1}{\rho},$$

where ρ = m/V - gas density.

Consequently, the specific volume and the density are mutually reciprocal values.

Kelvin's temperature T k as parameter of state of the gas is connected with the temperature of Celsius t°C by the dependence

$$T = t + 273{,}15 \approx t + 273.$$

To the standard conditions of state of the gas is related temperature of t = 0°C or, which is the same, T = 273 K.

Besides the indicated parameters gas is characterized by compressibility, temperature volumetric expansion coefficient, by ductility/toughness/viscosity and specific heat.

the specific (is volumetric) heat of gas. By the specific (volumetric) heat of gas is understood the ratio of the amount of heat, absorbed by the unit of the mass (volume) of gas, to the appropriate increase in the temperature. In this case are distinguished specific heat capacities at constant pressure c_p and at a constant volume c_V. The relationships between heat capacities with p = const and v = const are placed as the basis of gas dynamics. Specifically, the important parameter is the relation of these heat capacities

$$\frac{c_p}{c_V} = k,$$

called adiabatic index in the adiabatic process of a change in the

state of the gas.

The specific heat of imperfect gases depends on temperature, in connection with which they use the average for the assigned time interval of temperatures specific heat.

Viscosity of gas. The viscosity of gases usually is estimated at the value of dynamic viscosity μ. Unlike true liquids the dynamic viscosity of air with an increase in the temperature increases. Page 259.

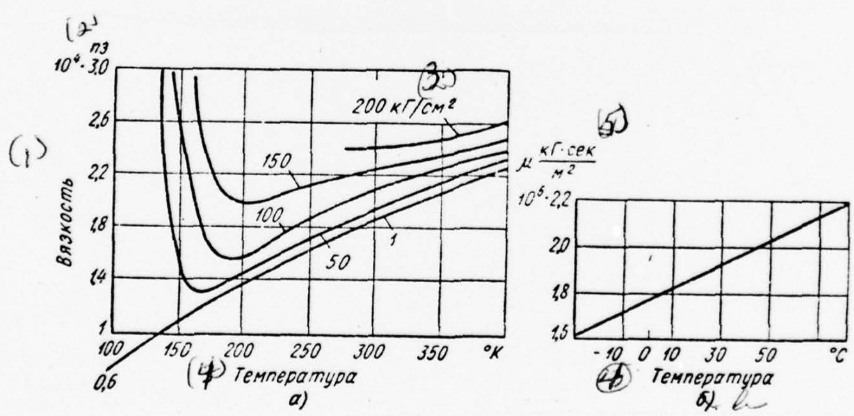

Fig. 221. Dependence of the ductility/toughness/viscosity of air on temperature.

Key: (1). Ductility/toughness/viscosity. (2). poise. (3). kgf/cm². (4). Temperature. (5). kg•cm²/m².

Table 2. Dynamic viscosity into 10^6 dyn•s/cm² [1].

FOOTNOTE [1]. 1 dyn•s/cm² = 0.1 Pa•s. ENDFOOTNOTE.

(1) p в кГ/см²	(2) t в °C			
	0	25	50	100
1	172,0	183,7	195,5	218,0
50	181,5	192,2	203,2	224,0
100	197,0	206,0	215,0	233,5

Key: (1). p in kgf/cm^2. (2). t in °C. The dependence of the ductility/toughness/viscosity of air on temperature sufficiently accurately is characterized by empirical formula (at constant pressure)

$$\mu=\mu_0\left(\frac{T}{273}\right)^{0,75},$$

where μ - the dynamic viscosity of air at the given temperature in K; μ_0 - the same with temperature of 0°C or 273 K; T is absolute temperature in K.

Temperature dependence of kinematic viscosity $\nu=\frac{\mu}{\rho}$

$$\nu=\nu_0\left(\frac{T}{273}\right)^{1,75}.$$

The viscosity of gases depends also on pressure, being raised with an increase in the latter (Table 2).

Figure 221a gives the graph/diagram of the dependence of the dynamic viscosity of air on temperature and pressure, while Fig. 221b - viscosity-temperature dependence at atmospheric pressure.

The dynamic viscosity of nitrogen at atmospheric pressure and temperature $t = 25°C$ is equal to $178 \cdot 10^{-6}$ dyn•s/cm² or 17.8 μPa•s.

Page 260.

The thermal expansion of gas. The thermal expansion of gas is characterized by the temperature volumetric expansion coefficient in $°C^{-1}$, that show a relative change in the volume of gas during a change in the temperature on 1°C:

$$\alpha = \frac{1}{V_0} \cdot \frac{\Delta V}{\Delta T},$$

where ΔV it is characterized a change of volume of V gas in m^3; ΔT - a change in the temperature in $^{\circ}C$ or K.

From thermodynamics it is known that the perfect gas is expanded at constant pressure (isobaric process is proportional to an increase in its absolute temperature of T (Gay-Lussac law). This law is described by the equation

$$v_T = v_0 (1 + \alpha t),$$

where v_T and v_0 - the specific volume of gas at the assigned and initial temperature; α is a temperature volumetric expansion coefficient of gas (virtually it can be accepted by constant for all gases); t - the temperature of gas in $^{\circ}C$.

If the volume of perfect gas is supported by constant (isochoric

process), then the pressure p_T in it grow/rises proportional to an increase in its absolute temperature:

$$p_T = p_0 (1 + \alpha t),$$

where p_0 it grow/rises the initial pressure.

For temperature of $t = 1/\alpha$, value p_T becomes equal to zero. This temperature, equal $t = 273,15°C$ or $T = °K$ is absolute zero. During the calculation/enumeration of temperature from absolute zero, it is called absolute temperature and is designated by T.

Compressibility of gas. The compressibility characterizes change ΔV the volume of gas during a change in the pressure on Δp:

$$\beta = \frac{1}{\Delta p} \cdot \frac{\Delta V}{V_0}.$$

EQUATION OF STATE OF PERFECT GAS.

The dependence of specific volume or density of gas on temperature of T and pressure p is called the equation of state of gas.

For a perfect gas (specifically, for air at relatively low pressures)

$$pv = RT. \tag{71}$$

This equation, obtain the name of characteristic equation or equation of state of gas, relates parameters p, v and T.

Taking into account that the specific volume

$$v = \frac{1}{\rho}, \tag{72}$$

equation of state or characteristic equation we can present in the form

$$\frac{p}{\rho}=RT, \qquad (73)$$

where R - the specific gas constant, equal for dry air 287.1 m^2/(s^2•°C) in the units of mk forces system, 287.1 J/ (kg•K) - in the units of SI and mks system; 29.27 kg•m/ (kg•°C) - in unity of two mk forces systems and of SI.

Page 261.

In the work of pneumatic actuators are possible the varied conditions of heat exchange between the driving under channels gas and the environment. At small rate of flow and during good heat exchange between the walls of channels (conduit/manifolds) and by the environment the processes, which take place within the limits of the volume elements of gas, can be close to isothermal (see page 273). The latter is confirmed by experiment and theoretical studies, which show that in the form of the flow resistance of gas (work of frictional forces is converted into heat) in long conduit/manifold process it occur/flow/lasts over isotherm ($p_2v_2 = p_1v_1 = const$).

However, in the general case a change of the state of the gas depending on the duration of process and specific environmentalconditions can occur/flow/last over different laws, with an arbitrary change in parameters p, v and T. In this case in all cases is satisfied the equation of state (71).

Such processes are the polytropic processes, characterized by the equation

$$p_2v_2^n = p_1v_1^n = \text{const},$$

where p_1 and p_2 - the initial and stagnation pressure; v_1 and v_2 are the initial and final specific volumes; n - polytropic exponent.

Polytropic exponent at the processes, utilized in technology, lie/rests usually at close margins. These limits are the described below special cases.

Isothermal process. This process is described by the equalities

$$T = \text{const}; \qquad n = 1.$$

According to Boyle's law of - Mariotte, the specific volume of gas inversely proportional to its pressure p

$$\frac{v_2}{v_1} = \frac{p_1}{p_2},$$

$$p_1 v_1 = p_2 v_2 = \text{const},$$

where p_1 and p_2 are the initial and final absolute pressures of gas; v_1 and v_2 is the specific volume of gas respectively at pressures p_1 and p_2; ρ - the density of gas (value, reciprocal to specific volume).

In the described process, which obtained name isothermal, gas it is compressed or it is expanded with the preservation/retention/maintaining of constant temperature.

It is obvious, that a similar process can occur only during a very slow change in the state (compression or expansion) of gas. This case includes, for example, the process of the discharge of the gas-hydraulic storage battery/accumulator of the hydraulic system of the press with the holding of workpiece under pressure (see Fig. 94).

Adiabatic process. Under the assumption that the process of a change in the state of the gas proceeds without heat exchange with the environment, let us have

$$pv^k = \text{const} \quad \text{or} \quad \frac{p}{\rho^k} = \text{const},$$

where p and ρ - the pressure and the density of gas; $k=\frac{c_p}{c_V}$ is an adiabatic index (for the dry air k = 1.405; here c_p and c_V is specific heat of gas at constant temperature and a constant volume.

The process, described by these equations, is called adiabatic.

In the practice of the process, close to adiabatic, is observed during the outflow of gas from the reservoir through the nozzle or the opening/aperture in the fine/thin wall, when as a consequence of the temporary stay of gas within the limits of nozzle it is possible to disregard frictional forces and heat exchange with the environment (with the walls of channel).

Page 262.

In the examination of the last/latter process it is necessary to keep in mind that thermal insulation is not, strictly speaking, on

the strength of frictional resistance during the flow of gas whose work is converted into heat, in a sufficient basis/base in order that the polytropic exponent it would be possible to take as equal to the index of isentrope k (by isentropic flow understand the heat-insulated flow of the perfect gas, in which there are no frictional forces). The latter can be correctly only for a perfect gas (or when it is possible to disregard frictional forces).

Specific volume v, pressure p and the absolute temperature T of perfect gas are connected in adiabatic process by the relationships

$$v_2 = v_1 (p_1/p_2)^{\frac{1}{k}};\quad v_2 = v_1 (T_1/T_2)^{k-1};$$

$$p_2 = p_1 (v_1/v_2)^k;\quad p_2 = p_1 (T_2/T_1)^{\frac{k}{k-1}};$$

$$T_2 = T_1 (v_1/v_2)^{k-1};\quad T_2 = T_1 (p_2/p_1)^{\frac{k-1}{k}}.$$

Polytropic process. Since under actual conditions during a change in the state of the gas unavoidably occurs certain heat exchange between gas and walls of container and the liquid, occurs the so-called polytropic change in the state of the gas, which is something average of the examined maximum changes (isothermal and adiabatic processes).

The equation for this state, which covers its all the possible in practice changes, takes the form

$$pv^n = \text{const} \text{ и } \frac{p}{\rho^n} = \text{const}, \qquad (74)$$

where k > n > 1 - polytropic exponent (with n = 1 we have isothermal, also, with n = k - adiabatic processes).

Accordingly, we have

$$p_1^{1/n} v_1 = p_2^{1/n} v_2.$$

The relationships of the parameters in polytropic process are expressed by equations for an adiabatic curve with the replacement of

coefficient k by coefficient of n.

If for the gas systems, in which are utilized the diatomic gases at low pressures (7-10 kgf/cm^2), polytropic exponent n of the processes of the emptying (and filling) of tank/bottles (capacitance/capacities) virtually it fluctuates between the indices of the isothermal and adiabatic processes $k > n > 1$, then by high-pressure systems (50-200 kgf/cm^2) it can exceed the adiabatic index of perfect gas $k = 1.4$. Thus, for instance, for imperfect gases, including for air, it can reach at temperatures from +100 to -60°C and pressure 50-100 kgf/cm^2 of value $n = 2$ and more [6]. This is explained by a change of the physical properties of gases in the range of the indicated pressures and temperatures as compared with the properties of perfect gases. It is obvious that the incorrect selection of polytropic exponent leads to considerable errors during the analysis of system, during the determination of pressures and temperatures and, in particular, at the emptying of tank/bottles. The latter especially is noticeable during the large expansions of gas (see page 265).

Polytropic exponent n can be determined only for concrete/specific/actual gases and conditions of their compression,

expansion and diversion/tap of heat (heat exchange).

Speeds of propagation of sound. In the theory of the flow of gases the important parameter is the speed of sound a, which is the velocity of propagation in the gaseous medium of slight disturbances. At the velocity of propagation in gas of acoustic wave is connected the rate of flow of gas, whereupon the rate of flow of gas, equal to the speed of sound, is the boundary, upon transition of which change the law governing gas flow the discharge characteristics of the pneumatic system.

Page 263.

Communication/connection of the speed of sound a with the properties of gases is expressed by the known artificial satellite of the course of gas dynamics by the equation

$$a=\sqrt{\frac{dp}{d\rho}},$$

where p and ρ - the pressure and the density of gas.

Under the assumption that the changes in the flow parameter of gas, caused by slight disturbances, occur so rapidly that it is possible to disregard heat exchange between the particles of gas, and disturbance/perturbations themselves, created by acoustic wave, are so small that it is possible to disregard frictional forces, we can write

$$\frac{dp}{d\rho} = k\frac{p}{\rho},$$

where k - adiabatic index.

Accordingly, the speed of sound can be expressed through the flow parameters of the gas:

$$a=\sqrt{k\frac{p}{\rho}}. \tag{74a}$$

For the perfect gas, for which is correct $p/\rho = RT$, the last/latter dependence can be rewritten in the form

$$a=\sqrt{kRT},$$

where R is specific gas constant.

FLOW OF GAS.

The engineering calculations of the flow of gas in the

cell/elements of pneumatic systems are reduced to the calculations, connected with the outflow of gas from reservoirs (tank/bottles) and with their filling, and also with flow on the conduit/manifolds of pneumatic systems and through the local friction.

These calculations on the strength of the compressibility of air present known difficulties, caused by the fact that its flow in the conduit/manifolds of pneumatic systems and the channels of their assemblies is accompanied, as this was shown, by a change in the pressure and the specific volume. In view of this during calculations they proceed from the condition that during the steady process of flow air mass flow m through any cross section of conduit/manifold with an area of f, remains constant, in accordance with how the mass flow rate is determined from the equation of the continuity (continuity) of the flow

$$m = fu\rho = \frac{uf}{v} = \text{const},$$

where f is a sectional area of gas flow (pipeline); u and ρ is the average speed and air density in this section; v is the specific

volume of air.

Since the volumetric flow rate

$$Q = uf = \frac{m}{\rho}$$

by way of air flow on conduit/manifold is not retained, but it increases as a result of the expansion, caused by decompression during flow according to expression (71), the average air speed along the length of conduit/manifold u = Q/f also will grow/rise. Page 264.

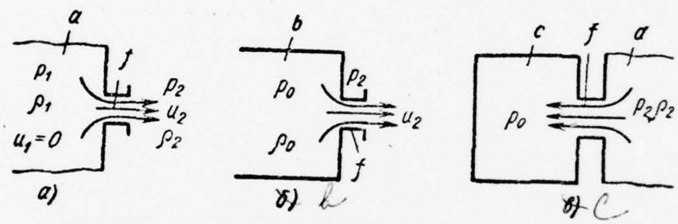

Fig. 222. The design diagrams of the outflow of gas. In this case as a result of the expansion of air occurs also a change in its temperature, that also must be taken into account during calculations.

As the basis of the calculations of the flow of gas in the cell/elements of pneumatic systems is placed known from the course "gas hydrodynamics" the equation of motion of perfect gas in the adiabatic mode/conditions

$$\frac{u_1^2}{2}+\frac{k}{k-1}\cdot\frac{p_1}{\rho_1}+gz_1=\frac{u_2^2}{2}+\frac{k}{k-1}\cdot\frac{p_2}{\rho_2}+gz_2,$$

where ρ_1 and ρ_2; p_1 and p_2; u_1 and u_2 - respectively density, the pressure and the gas velocity in the initial and final sections of gas flow in question; k is an adiabatic index; z_1 and z_2 are leveling height/altitudes of the points of gas flow in question.

By disregarding a difference in the leveling height/altitudes $z_1 - z_2$, we will obtain

$$\frac{u_1^2}{2}+\frac{k}{k-1}\cdot\frac{p_1}{\rho_1}=\frac{u_2^2}{2}+\frac{k}{k-1}\cdot\frac{p_2}{\rho_2}. \quad (75)$$

Outflow of gas from the reservoir of the unlimited capacitance/capacity. The calculations of the outflow of gases (air) from the reservoirs of the unlimited and limited capacitance/capacity and filling of the latter are basic in the calculations of pneumatic systems. The outflow of gas in the general case occurs with the discharge of hydroaccumulators, with the emptying of pneumatic cylinders and etc.

Accepting in equation (75) $u_1 = 0$ (i.e. disregarding the gas velocity in feed tank), we find the calculated exhaust gas velocity $u_2 = u$ from reservoir a of the unlimited (infinite large) capacitance/capacity (Fig. 222a) through the round opening/aperture (or nozzle) in wall during the adiabatic process:

$$u=\sqrt{\frac{2k}{k-1}\left(\frac{p_1}{\rho_1}-\frac{p_2}{\rho_2}\right)}. \qquad (76)$$

For the flow of perfect gas we have

$$\frac{p_2}{\rho_2^k} = \frac{p_1}{\rho_1^k} \text{ и } \rho_2 = \rho_1 \left(\frac{p_2}{p_1}\right)^{\frac{1}{k}}. \quad (77)$$

After substituting these values into equation (76) and by converting, we will obtain

$$u = \sqrt{\frac{2k}{k-1} \cdot \frac{p_1}{\rho_1} \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}, \quad (78)$$

where k - adiabatic index; p_2/p_1 - the ratio of the pressure of gas, here p_1 and p_2 are a gas pressure in reservoir and in the medium, into which occurs the discharge (pressure in the final and initial section of gas flow).

Page 265.

Mass flow rate m of the gas, which takes place at a rate of u through the opening/aperture with an area of f, kg/s:

$$m = Q\rho_2 = uf\rho_2 = f\rho_2 \sqrt{\frac{2k}{k-1} \cdot \frac{p_1}{\rho_1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}, \qquad (79)$$

where Q = uf - the volumetric flow rate of gas.

Taking into account expression (77) the last/latter equation can be rewritten in the form

$$m = f\sqrt{\frac{2k}{k-1}p_1\rho_1}\sqrt{\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}} = \psi f\sqrt{p_1\rho_1}, \qquad (80)$$

where

$$\psi = \sqrt{\frac{2k}{k-1}}\sqrt{\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}}.$$

After designating the expansion ratio of the gas by

$$\frac{p_2}{p_1} = x, \tag{81}$$

equation (80) we can present in the form

$$m = f\sqrt{\frac{2k}{k-1}\, p_1\rho_1}\sqrt{x^{\frac{2}{k}} - x^{\frac{k+1}{k}}}, \tag{82}$$

or, after introducing gas constant R,

$$m = f\sqrt{2\frac{p_1^2}{RT_a}\cdot\frac{k}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]} =$$
$$= fp_1\sqrt{\frac{2}{RT_a}\cdot\frac{k}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]}, \qquad (83)$$

where T_a is temperature of gas at the inlet into the throttling opening/aperture into K.

The given calculations are produced without taking into account of the losses, caused by friction, heat exchange and other factors. The account of these losses usually they produce, accepting, that the process proceeds under the polytropic conditions. In this case the gas flow is calculated from expression (81) with the replacement of adiabatic index k by polytropic exponent n whose value is taken as equal by n = 1.3-1.35.

Losses to friction are considered frequently by the coefficient of expenditure/consumption μ, introducing it into given formula (81) for an adiabatic process.

As a result obtain formula for the mass flow rate taking into account friction

$$m' = \mu f \sqrt{\frac{2k}{k-1} p_1 \rho_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1} \right)^{\frac{k+1}{k}} \right]}, \tag{84}$$

where $\mu = \frac{1}{\sqrt{1+\zeta}} < 1$ - the practical coefficient of expenditure/consumption (experiments show that the value μ can be calculated with sufficient accuracy/precision in formulas for the incompressible fluid); ζ is a coefficient of the local friction.

Critical expansion of gas. The maximum mass calculated flow rate of gas corresponds to the condition of the equality of zero

derivative under equation (80).

Page 266.

The maximum calculated flow rate corresponds to the critical expansion of gas (to critical ratio of pressure)

$$\frac{p_2}{p_1} = x_{\kappa p},$$

in which discharge velocity according to equation (78) becomes equal to the speed of sound in gas at the parameters of the latter, that correspond to the parameters at output/yield from the throttling nozzle [cm. equation (77)].

The parameters of the critical expansion, during which occurs the greatest flow rate, we will obtain by the investigation of the function

$$\psi(x,\ k)=\sqrt{\frac{k}{k-1}\left(x^{\frac{2}{k}}-x^{\frac{k+1}{k}}\right)}$$

for maximum, as a result of which let us have

$$\left(\frac{p_2}{p_1}\right)_{\kappa p}=x_{\kappa p}=\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}. \qquad (85)$$

For an adiabatic process $x_{\kappa p}=0{,}528$. This condition corresponds to rate of flow, equal to the local speed of propagation of sound.

Given theoretically calculations of discharge through the opening/aperture under the assumption of adiabatic mode/conditions can be with sufficient accuracy/precision applied also for practical

calculations of discharge through the short nozzle, with which it is possible to disregard frictional forces, and also as a result of the short-term determination of gas in nozzle to disregard heat exchange with the environment.

The analysis of the function

$$\psi(x, k) = \sqrt{\frac{k}{k-2}\left(x^{\frac{2}{k}} - x^{\frac{k+1}{k}}\right)}$$

shows [see also equality (85)], that there are two zones (range) of the flow:

the zone, which corresponds to the examined above condition $x_{кр} < x < 1$, which is called of the zone of subcritical (subcritical) flow (gas velocity in this zone lower than the speed of sound);

the zone, which corresponds to condition $0 < x < x_{кр}$, which is called of the zone of supercritical flow (gas velocity in this zone

is constant and close or equal to the speed of sound).

Accordingly, are distinguished the processes, which take place in subcritical (lower than critical) and supercritical mode/conditions.

In supercritical zone occurs the maximum and constant mass flow rate, which corresponds to the critical expansion of gas. Formula for determining the flow rate in this zone we will obtain, after substituting into equation (80) the value of the critical expansion of the gas

$$\left(\frac{p_2}{p_1}\right)_{\kappa p} = x_{\kappa p} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}.$$

As a result we will obtain

$$m_{\max}=f\sqrt{\frac{2k}{k-1}}\sqrt{p_1\rho_1}\sqrt{\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}\cdot\frac{2}{k}}-\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}\cdot\frac{k+1}{k}}}=$$

$$=f\left(\frac{2}{k+1}\right)^{\frac{1}{k-1}}\sqrt{\frac{2k}{k+1}p_1\rho_1}, \qquad (86)$$

where ρ_1 - gas density before the exit nozzle (throttle/choke), expressed as the initial at the torque/moment of the beginning of discharge parameters.

[redacted]

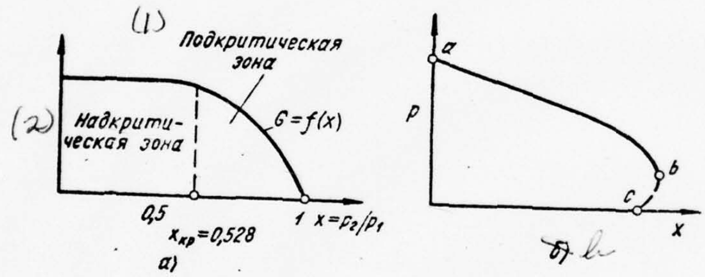

Fig. 223. Characteristics of the outflow of gas from tank/bottle (a) and flows in conduit/manifold (b) depending on expansion ratio x.

Key: (1). Subcritical zone. (2). Supercritical zone. Taking into account losses to friction

$$\dot{m}_{max} = \mu f \left(\frac{2}{k+1}\right)^{\frac{1}{k-1}} \sqrt{\frac{2k}{k+1} p_1 \rho_1}, \qquad (87)$$

where μ is a coefficient of flow rate (see above).

The curve/graph of a change in flow rate of m, deposited along the axis of ordinates, in the function

$$x = \frac{p_2}{p_1}$$

is shown in Fig. 223a. In supercritical zone ($x < 0.528$) we have the constant and maximum mass flow rate, in subcritical - the alternating/variable flow rate, which decreases with increase in x.

Using the given equations, let us show that the gas flow during

its discharge at the speed of propagation of sound in gas (it corresponds $u_{\kappa p}$) it is maximum.

It is obvious, to the maximum flow rate, expressed by equation (86), corresponds the maximum (critical) exhaust gas velocity according to equation (78):

$$u_{2\,max} = u_{\kappa p} = \sqrt{\frac{2k}{k-1}\cdot\frac{p_1}{\rho_1}\left(1 - x_{\kappa p}^{\frac{k-1}{k}}\right)}$$

or taking into account equation (85)

$$u_{\kappa p} = \sqrt{\frac{2k}{k+1}\cdot\frac{p_1}{\rho_1}}.$$

By converting equations (85) and (77), we will obtain

$$p_1 = p_{2\,max}\left(\frac{2}{k+1}\right)^{-\frac{k}{k-1}} \text{ and } \rho_1 = \rho_{2\,max}\left(\frac{p_1}{p_{2\,max}}\right)^{\frac{1}{k}} = \rho_{2\,max}\left(\frac{2}{k+1}\right)^{-\frac{1}{k-1}},$$

and then critical discharge velocity

$$u_{\kappa p} = \sqrt{k\frac{p_{2\,max}}{\rho_{2\,max}}}. \qquad (88)$$

From the comparison of equations (88) and (74a) it follows that the maximum (critical) gas velocity (and, consequently, its maximum mass flow rate) occurs at the speed of sound in the gas

$$a = \sqrt{k \frac{p_{max}}{\rho_{max}}}.$$

Emptying of the reservoirs of the limited capacitance/capacity. In engineering practice in essence it is necessary to perform the calculations, connected with emptying and tankage (tank/bottles) of the limited capacitance/capacity.

Page 268.

These cases include the emptying of gas cylinders in the process of the power supply of pneumatic systems, the filling or the emptying of pneumatic cylinders and etc.

The outflow of gas from the reservoir of the limited capacitance/capacity is characterized by the fact that at the limited tankage the parameters of the escaping gas will be time varying.

Let us examine the process of the emptying of cylinder b with a volume of V_0 through the opening/aperture with an area of f (Fig. 222b). The differential equation of the outflow of gas from this cylinder they compose on the basis of the condition that mass m of the gas, which takes place through the assigned cross section of opening/aperture for certain cut of time, is equal to a change of mass $dm = V_0 \, d\rho_i$ (where V_0 - the volume of cylinder, ρ_i - the instantaneous value of gas density) gas in cylinder for the same time.

Let in certain interval of time from the torque/moment of discovery/opening the opening/aperture (nozzle) of the emptyd cylinder the absolute storage pressure was equal to p_i and gas density in it ρ_i. The elementary mass dm gas, the past through this opening/aperture with an area of f for the interval of time dt, is equal according to equation (80)

$$dm = m \, dt = \psi f \sqrt{p_i \rho_i} \, dt, \qquad (89)$$

where p_i and ρ_i - the instantaneous values of pressure and density of gas in cylinder.

Expressing the current parameters of gas p_i and ρ_i through their initial values p_0 and ρ_0, that occurred in the cylinder before beginning of discharge, and assuming that a change in these parameters within cylinder at its emptying (outflow of gas) is subordinated to certain polytropic dependence with polytropic exponent n, we will obtain

$$\frac{p_i}{\rho_i^n} = \frac{p_0}{\rho_0^n}; \quad \rho_i = \rho_0 \left(\frac{p_i}{p_0}\right)^{\frac{1}{n}}. \tag{90}$$

After substituting equation (90) into equation (89), we will obtain

$$dm = \psi f \sqrt{p_i \rho_0 \left(\frac{p_i}{p_0}\right)^{\frac{1}{n}}} dt = \psi f \sqrt{p_0 \rho_0} \sqrt{\left(\frac{p_i}{p_0}\right)^{\frac{n+1}{n}}} dt. \quad (91)$$

Let us compose now expression for a change of the mass of gas in cylinder for the same interval of time dt. This mass at the moment of time t is equal to $m = V_0 \rho_i$, and consequently,

$$dm = V_0 \, d\rho_i,$$

where v_0 is equal the volume of cylinder.

Taking into account equation (90) we will obtain

$$dm = V_0 \rho_0 d\left[\left(\frac{p_i}{p_0}\right)^{\frac{1}{n}}\right] = \frac{V_0}{n} \rho \left(\frac{p_i}{p_0}\right)^{\frac{1}{n}-1} d\left(\frac{p_i}{p_0}\right). \quad (92)$$

After equating equations (91) and (92), we will obtain (taking into account sign) the differential equation of the emptying of the cylinder of the limited capacitance/capacity V_0

$$\frac{1}{n} V_0\rho_0\left(\frac{p_i}{p_0}\right)^{\frac{1}{n}-1} d\left(\frac{p_i}{p_0}\right) = -\psi f \sqrt{p_0\rho_0} \sqrt{\left(\frac{p_i}{p_0}\right)^{\frac{n+1}{n}}}\, dt.$$

After reducing on $\left(\frac{p_i}{p_0}\right)^{\frac{1}{n}}$, we will obtain

$$\frac{1}{n} V_0 \rho_0 \left(\frac{p_i}{p_0}\right)^{-1} d\left(\frac{p_i}{p_0}\right) = -\psi f \sqrt{p_0 \rho_0} \left(\frac{p_i}{p_0}\right)^{\frac{1}{2} - \frac{1}{2n}} dt.$$

For the integration of this equation (with f = const) let us produce the conversion:

$$\frac{1}{n} \frac{1}{\left(\frac{p_i}{p_0}\right)\left(\frac{p_i}{p_0}\right)^{\frac{1}{2} - \frac{1}{2n}}} d\left(\frac{p_i}{p_0}\right) = -\psi \frac{f}{V_0} \sqrt{\frac{p_0}{\rho_0}}\, dt$$

or

$$\frac{1}{n} \left(\frac{p_i}{p_0}\right)^{\frac{1}{2n} - \frac{3}{2}} d\left(\frac{p_i}{p_0}\right) = -\psi \frac{f}{V_0} \sqrt{\frac{p_0}{\rho_0}}\, dt. \qquad (93)$$

Page 269.

During the integration of equation (93) we observe the same two cases (two zones) of the discharge:

a) the discharge in the supercritical zone, when $x_{\kappa p} > \frac{p_2}{p_i} > 0$ (in this case $\psi = \text{const} = \psi_{max}$ and the integration of differential equation (93) it presents no difficulties);

b) the discharge in the subcritical zone, when $1 > \frac{p_2}{p_i} > x_{\kappa p}$, in this case $\psi = \text{var}$ (in view of the complexity of integrand the

integration of equation (93) conduct by grapho-analytic method).

Discharge in supercritical zone. During calculation in this case values $\psi = \psi_{max}$ [see formula (80)] we substitute $p_2 = p_{\kappa p}$:

$$\frac{p_2}{p_i} = \frac{p_{\kappa p}}{p_i} = x_{\kappa p} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}},$$

where p_i and p_2 are the current storage pressure and in the environments; $p_{\kappa p}$ - critical pressure in jet.

As a result

$$\psi_{max} = \sqrt{\frac{2k}{k-1}\left[\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}\cdot\frac{2}{k}} - \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}\cdot\frac{k+1}{k}}\right]}.$$

After conversions we will obtain

$$\psi_{max}=\left(\frac{2}{k+1}\right)^{\frac{1}{k-1}}\sqrt{\frac{2k}{k+1}}.$$

After substituting ψ_{max} into differential equation (93), we will obtain

$$\frac{1}{n}\left(\frac{p_i}{p_0}\right)^{\frac{1}{2n}-\frac{3}{2}} d\left(\frac{p_i}{p_0}\right) = -\psi_{max}\frac{f}{V_0}\sqrt{\frac{p_0}{\rho_0}}\,dt.$$

Integrating within limits from p_0 to p_i and from t = 0 to t, we have

$$\frac{1}{n}\int_1^{p_i/p_0}\left(\frac{p_i}{p_0}\right)^{\frac{1}{2n}-\frac{3}{2}} d\left(\frac{p_i}{p_0}\right) = -\psi_{max}\frac{f}{V_0}\sqrt{\frac{p_0}{\rho_0}}\int_0^t dt.$$

By converting, we will obtain equation for the calculation of time t of the partial emptying of cylinder for the assigned condition of decompression under cylinder from the initial p_0 to that which was

assigned p_i for the mode/conditions, which corresponds to the supercritical range, when $p_0 > p_i \geqslant p_2/x_{\kappa p}$:

$$t = \frac{2}{n-1} \cdot \frac{V_0}{\varphi_{\max} f} \sqrt{\frac{p_0}{\rho_0}} \left[\left(\frac{p_0}{p_i} \right)^{\frac{n-1}{2n}} - 1 \right]. \qquad (94)$$

Complete time of the discharge in supercritical range, which corresponds to decompression from p_0 to $\frac{p_2}{x_{\kappa p}}$, is calculated by means of substitution into equation (94)

$$p_i = \frac{p_2}{x_{\kappa p}},$$

as a result of which after conversions we will obtain

$$t_{\kappa p}=\frac{2}{n-1}\cdot\frac{V_0}{\psi_{max}}\cdot\frac{\sqrt{\rho_0}}{f\sqrt{p_0}}\left[\left(\frac{p_0}{p_2}\right)^{\frac{n-1}{2n}}\left(\frac{2}{k+1}\right)^{\frac{n}{n-1}\cdot\frac{n-1}{n}}-1\right].$$

Page 270.

In a similar manner we can obtain the differential equation of the discharge in subcritical range for decompression to p_i

$$t=\frac{1}{n}\cdot\frac{V_0}{f}\sqrt{\frac{\rho_0}{p_0}}\left(\frac{p_0}{p_2}\right)^{\frac{1}{2}-\frac{1}{2n}}\int_{p_0}^{p_i}\frac{1}{\left(\frac{p_2}{p_i}\right)^{\frac{1}{2n}+\frac{1}{2}}\psi}\,d\left(\frac{p_i}{p_2}\right).$$

In view of the complexity of function this equation usually is solved grapho-analytically and here it is not examined.

Tankage of the limited capacitance/capacity. The cases of the tankage of the limited capacitance/capacity include the filling of the pneumatic cylinders of pneumatic systems, the charge of gas-hydraulic storage battery/accumulators and etc. With the filling with the compressed air of any capacitance/capacity air at the initial moment, when pressure in the filled capacitance/capacity minimum, flows, being expanded at the maximum speed, which with

pressure balance in supply main and the filled capacitance/capacity is reduced, reaching during the complete equalization of these pressures of zero value. It is obvious, in this case will be variable the consequence of the expansion of air and its temperature, the changes in it can occur over a wide range.

Let to capacitance/capacity d with the unlimited volume and the constant pressure p_2 be connected the capacitance/capacity (cylinder) c with a volume of V_0 and pressure $p_0 < p_2$ (Fig. 223). In this case we assume that the volume of the source of flow rate d is so great that by a change in the pressure and by a change in the speed of the displacement/movement of particles of the gas in it with filling of capacitance/capacity c it is possible to disregard.

Let us determine time of a pressure increase in the connected cylinder with p_0 to $p_1 = p_2$, i.e., let us determine time of pressure balance between the source of flow rate and the filled cylinder.

Let at the moment of time t the storage pressure will be p_1, gas density in jet ρ and velocity u. Mass dm gas, that flows into the cylinder through the opening/aperture of constant section f for time

dt, will compose

$$dm = \rho u f\,dt = f\rho\sqrt{\frac{2k}{k-1}\cdot\frac{p_2}{\rho_2}\left[1-\frac{p_i}{p_2}^{\frac{k-1}{k}}\right]}\,dt, \tag{95}$$

where u is the gas velocity, determined according to equation (78).

Solving differential equation (95) taking into account velocity u and the functions

$$x=\frac{p_i}{p_2} \text{ and } \frac{p_2}{\rho_2}=RT_2,$$

we will obtain equation for the elementary mass dm gas, through cross section f the jet taking place for the segment element of time dt:

$$dm = fp_2\sqrt{\frac{\rho_2}{p_2}}\,\psi(x)\,dt = fp_2\sqrt{\frac{1}{RT_2}}\,\psi(x)\,dt, \tag{96}$$

where T_2 is the absolute temperature of gas, calculated according to characteristic equation (73).

Let us compose now equations for a change of the mass of gas in reservoir for the same time interval dt:

$$dm = V_0 d\rho_i, \tag{97}$$

where $m = V_0\rho_i$ is a mass of gas in reservoir with a volume of V_0 at the moment of time t; $\rho_i = \frac{p_i}{RT_i}$ - the instantaneous value of gas density; here R is specific gas constant.

Page 271.

Investigations show that the process of tankage (cylinders)

varies from adiabatic in the beginning filling to isothermal at the end filling, as a result the temperature in reservoir gradually it approaches from T_i T_2. By set/assuming $T_i \approx T_2$, we will obtain

$$\rho_i = \frac{p_i}{RT_2}.$$

In this case

$$d\rho_i = \frac{dp_i}{RT_2}.$$

Taking into account the last/latter equality equation (97) will take the form

$$dm = \frac{V_0}{RT_2} dp_i. \tag{98}$$

After equating the right sides of equations (96) and (98), we will obtain

$$\frac{fp_2}{\sqrt{RT_2}} \cdot \psi(x)\, dt = \frac{V_0}{RT_2} dp_i,$$

whence

$$dt = \frac{V_0}{fp_2 \sqrt{RT_2}} \cdot \frac{dp_i}{\psi(x)}.$$

Taking into account that

$$\frac{dp_i}{p_2} = dx,$$

we obtain

$$dt = \frac{V_0}{f\sqrt{RT_2}} \cdot \frac{dx}{\psi(x)}.$$

Integrating this equation within the limits

$$x_0 = \frac{p_0}{p_2} \text{ до } x_i = \frac{p_i}{p_2},$$

we will obtain taking into account the character of the process (super- or subcritical) taking place the unknown opening time:

a) in supercritical range (x_0 and $x_i < x_{кр} = 0{,}528$, and also $\psi(x) = \psi_{max}$)

$$t=\frac{V_0}{f_0\sqrt{RT_2}}(x_i-x_0);$$

b) in subcritical range (x_0 and $x_i > x_{\kappa p} > 0{,}528$)

$$t=\sqrt{\frac{2k}{k-1}}\cdot\frac{V_0}{f\sqrt{RT_2}}\left[\sqrt{1-x_0^{\frac{k-1}{k}}}-\sqrt{1-x_i^{\frac{k-1}{k}}}\right].$$

Actually discharge will occur not on adiabatic, but on polytropic cycle, in view of which for the calculations necessary to know polytropic exponent. Experiment shows that if the emptying (either filling) of capacitance/capacity occurs through the opening/aperture (throttle/choke) or the short branch pipe, with which noticeable heat exchange with environment does not occur, polytropic exponent will be close to adiabatic index $n \approx k$ and

$$\frac{p_2}{p_1}=0{,}528.$$

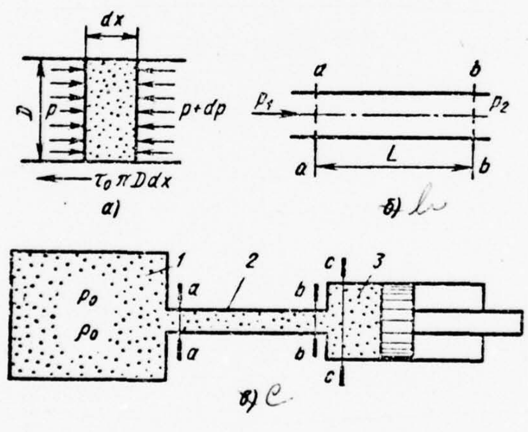

Fig. 224. The design diagrams of the flow of gas in conduit/manifold.

¶With a decrease in the polytropic exponent n value p_2/p_1 will

increase, reaching with n = 1

$$\frac{p_2}{p_1} = 0{,}607.$$

Flow of gas in conduit/manifold. Important for a practice is also the calculation of air flow (gases) in conduit/manifold.

For a derivation of the differential equation of the steady flow of gas in conduit/manifold let us isolate segment element with an its length of dx (Fig. 224a) and, after using to the volume element of gas dV = fdx, shown by point shading, the equation of momentum (nonuniformity of the distribution of velocities over the section of conduit/manifold we disregard), will write

$$p\frac{\pi D^2}{4}-(p+$$
$$+dp)\frac{\pi D^2}{4}-\tau_0\pi D\,dx=$$
$$=\frac{\pi D^2}{4}\cdot\frac{d\beta\rho u^2}{dx}dx, \qquad (99)$$

where D - the diameter of the internal section of conduit/manifold; $\tau_0 = \lambda\ \rho u^2/8$ - the tangential shear stress of the layers of liquid; p and dp - pressure in this section of conduit/manifold and elementary change in it at length dx; β is the coefficient, which characterizes the nonuniformity of the distribution of velocities over the section of conduit/manifold; u is the average over the section of conduit/manifold gas velocity.

We convert the right side of the equation:

$$\frac{\pi D^2}{4}\cdot\frac{d}{dx}(\rho\beta u^2)\,dx=\frac{\pi D^2}{4}\cdot\frac{d}{dx}[(\rho\beta u)\,u]\,dx=$$
$$=\frac{\pi D^2}{4}\left[\beta\rho u\,\frac{du}{dx}+u^2\rho\,\frac{d\beta}{dx}+u\beta\,\frac{d\rho u}{dx}\right].\qquad(100)$$

Taking into account the continuity

$$\frac{d\rho u}{dx}=0$$

equation (99) assumes the form

$$-\frac{dp}{dx}=\frac{4}{D}\tau_0+\rho u^2\frac{d\beta}{dx}+\beta\rho u\frac{du}{dx}.$$

For turbulent flow conditions, during which the average speed of flow and the pressure in its each of sections are retained virtually constants (it corresponds to the establish/installed flow conditions), it is possible to accept $\beta = 1$ and respectively

$$\frac{d\beta}{dx} = 0.$$

Page 273.

As a result we will obtain

$$-\frac{dp}{dx}=\frac{4\tau_0}{D}+\rho u\frac{du}{dx}.$$

After substituting tangential shear stress of the layers of the liquid

$$\tau_0=\lambda\frac{\rho u^2}{8},$$

we will obtain the differential gas-flow equation in the conduit/manifold

$$\frac{dp}{\frac{\rho u^2}{2}} + \lambda \frac{dx}{D} + 2 \frac{du}{u} = 0, \tag{101}$$

where λ is a coefficient of friction drag [this coefficient can be calculated from expression (17) under the condition of the substitution of the average values of the entering it parameters].

After integrating the given differential equation taking into account the assigned gas process, we will obtain the equation of the steady flow of gas in conduit/manifold taking into account friction.

The calculations and experiment show that as a result of heat exchange the flow of gas can be close to adiabatic only with the very short cuts of conduit/manifolds (in the local friction) and with large pressure differentials (expansions of gas). With long conduit/manifolds this process under normal conditions more close to isothermal, while at the known length of conduit/manifolds it is isothermal, i.e., the temperature of gas in this case is retained

virtually constant over entire length of conduit/manifold.
Accordingly with long conduit/manifolds, and in particular with small pressure differentials, is valid the isothermal dependence

$$\frac{p_1}{\rho_1}=\frac{p}{\rho},$$

where p_1 and ρ_i; p and ρ - respectively absolute pressure and gas density in the initial section (a-a) and in arbitrary section (b-b) (Fig. 224b).

Furthermore, from the flow-continuity condition

$$u=u_1\frac{\rho_1}{\rho},$$

where u_1 and u are velocities in sections a-a and b-b.

Taking into account also that $\rho_1 = \rho\, p_1/p$, we have $u = u_1\, p_1/p$, or differentially

$$\frac{du}{u} + \frac{dp}{p} = 0.$$

Accordingly, the first term of equation (101) will take the form

$$\frac{2}{\rho_1 p_1 u_1} p\, dp$$

Furthermore, since

$$\mathrm{Re} = \frac{u\,d\rho}{\mu}.$$

in the process in question virtually it does not change along the length of conduit/manifold ($u\rho$ = const, μ = const, the relative roughness of duct k/d = const), then constant along the length of conduit/manifold it will be also the coefficient of friction λ.

Page 274.

Taking this into account integration of differential equation (101) along the length L of the cut of conduit/manifold will give (Fig. 224b)

$$p_1^2 - p_2^2 = \rho_1 u_1^2 p_1 \left(\lambda \frac{L}{D} - 2 \ln \frac{p_2}{p_1}\right). \qquad (102)$$

Since log term in the brackets of the last/latter equation is low in comparison with λ L/D, this term usually they disregard. As a result we will obtain the simplified expression

$$p_1^2 - p_2^2 = \lambda \frac{L}{D} \rho_1 u_1^2 p_1.$$

From this equation it follows that during the flow of gases a pressure drop along the length of conduit/manifold is expressed by

the power dependency, but not linear, as it takes place during flow of liquid, that it is evident from equation (16).

After introducing into equation (102) number Maxa.

$$M_1=\frac{u_1}{a_1},$$

where $a_1=\sqrt{k\frac{p_1}{\rho_1}}$ is the local speed of sound (see page 262), we will obtain

$$p_1^2-p_2^2=kM_1^2p_1^2\left(\lambda\frac{L}{D}-2\ln\frac{\rho_2}{\rho_1}\right), \qquad (103)$$

where k - adiabatic index.

Equation (103) makes it possible to calculate pressure in conduit/manifold at the required distance L from the initial

(initial) section, for which assigned/prescribed number Maxa.

By solving equation (103) relatively λ L/D, we will obtain

$$\lambda \frac{L}{D} = \frac{1}{kM_1^2}\left[1 - \left(\frac{p_2}{p_1}\right)^2\right] + 2\ln\frac{p_2}{p_1}. \qquad (104)$$

The last/latter equation shows a change of the dimensionless length L/D conduit/manifold in the function of the ratio of the pressure of gas p_2/p_1.

By disregarding log term, equation (104) can be presented in the form

$$\lambda \frac{L}{D} = \frac{1}{kM_1^2}\left[1-\left(\frac{p_2}{p_1}\right)^2\right]$$

or

$$\lambda = \frac{D}{LkM_1^2}\left[1-\left(\frac{p_2}{p_1}\right)^2\right].$$

Taking into account number Maxa to differential equation (101) we can give another form, after substituting

$$\frac{du}{u} = -\frac{dp}{p}.$$

In this case we will obtain

$$\frac{dp}{dx}=\frac{p\frac{\lambda}{2D}}{1-\frac{p}{\rho}u^2}=\frac{\frac{\lambda}{D}\cdot\frac{\rho u^2}{2}}{kM^2-1}. \qquad (105)$$

The common character of a change in pressure p along the length of conduit/manifold is shown in Fig. 223b. The flow of gas can exist only on the section of curve a-b. Segment of the curve b-c corresponds to supersonic flow. Point b is called limit point. Number

Maxa for the state of the flow of gas at this point is called the maximum number Maxa M_{np}.

Page 275.

With respect to this there are a maximum (low) pressure p_{np} and a limiting maximum length L_{np}.

Since at limit point (section of conduit/manifold) b

$$\frac{dp}{dx} = \infty,$$

the denominator of equation (105) will be equal to zero, and the maximum number Maxa

$$M_{np} = \frac{1}{\sqrt{k}}.$$

Consequently, the air speed in conduit/manifold can grow/rise only until number Maxa achieves in the maximum section, which must be found at the end of the conduit/manifold, value

$$M_{np} = \frac{1}{\sqrt{k}} = 0{,}845,$$

where $k = \frac{\sqrt{a}}{RT}$ - adiabatic index.

For the determination of the minimum ultimate pressure in turboconductor, which corresponds to this condition, we will use the

equality

$$u_1 p_1 = up = u_{np} p_{np},$$

taking into account which we will obtain

$$\frac{p_{np}}{p_1} = \frac{u_1}{u_{np}} = M_1 a_1,$$

whence

$$p_{np} = p_1 M_1 \sqrt{k}.$$

After substituting from the latter equation p_{np} into equation (104), we will obtain

$$\frac{\lambda L_{np}}{D}=\frac{1}{kM_1^2}-\left(1+\ln\frac{1}{kM_1^2}\right)$$

or

$$L_{np}=\frac{D}{\lambda}\left[\frac{1}{kM_1^2}-\left(1+\ln\frac{1}{kM_1^2}\right)\right].$$

Taking into account the equality

$$\frac{p_2}{p_1}=\frac{M_1}{M_2}$$

we can obtain

$$\frac{\lambda L}{D} = \left(\frac{\lambda L_{np}}{D}\right)^{M1} - \left(\frac{\lambda L_{np}}{D}\right)^{M2}.$$

From the given analysis it follows that the rate of flow of gas (and the number Maxa) grow/rises to the greatest (maximum) value $\left(M_{np} = \frac{1}{\sqrt{k}}\right)$ at certain limit point, which must be at the end of the conduit/manifold

The maximum gas flow during flow in the mode/conditions

$$\frac{p_2}{\rho_2^k} = \frac{p_1}{\rho_1^k}$$

occurs with M = 1. With M > 1, the pressure in flow along the length

of conduit/manifold and the speed of flow do not depend on pressure at the end of the conduit/manifold. In this case the frictional forces brake flow (shock wave), in view of which flow at the supersonic speed possibly only within the limits of the determined (critical) length of conduit/manifold.

Page 276.

If the length of conduit/manifold exceeds this value, then in certain of its section appears shock wave, as a result of which the supersonic speed abruptly pass into subsonic. After this shock wave the character of the flow of gas will change: speed along conduit/manifold again increases, and the pressure and the density of gas decrease.

The asymptotic relation of pressures (expansion of gas) p_2/p_1 and maximum length L_{np} they depend only on adiabatic index k (from the initial number Maxa). Respectively minimum ultimate pressure p_{np} in conduit/manifold and the maximum (greatest) length of conduit/manifold L_{np} can be expressed thus:

$$p_{np} = p_1 M \sqrt{k}; \quad L_{np} = \frac{D}{\lambda}\left[\frac{1}{kM_1^2} - \left(1 + \ln\frac{1}{kM_1^2}\right)\right].$$

Virtually in the conduit/manifolds of real lengths critical expansion ratio $x_{\kappa i}$ is not reached, i.e., the flow of gas on conduit/manifolds by length greater than L_{np} occurs in the zone of subcritical flow conditions.

Filling of pneumatic cylinder by the gas through the long conduit/manifold. Let us examine the diagram of the supply of air to pneumo-capacitance/capacity, shown in Fig. 224c. Air from air collector (receiver) 1 is headed on conduit/manifold 2 for pneumatic cylinder 3.

During the flow of gas according to this diagram it is possible

to isolate three sections: 1) discharge from air collector into conduit/manifold; 2) flow on conduit/manifold; 3) discharge from conduit/manifold into cylinder. With respect to this must be comprised the equations, which give the system, which is subject to solution. Problem is obtained complexly, in view of which let us examine approximate solution with the following assumptions: pressure upon the inlet into conduit/manifold (section a-a) equal to pressure p_0 in air collector and flow in conduit/manifold isothermal.

After designating by m air mass flow, we will obtain:

1) the gas velocity in section a-a according to equation (79)

$$u_a = \frac{m}{f\rho_a}, \qquad (106)$$

where f is a sectional area of conduit/manifold; ρ_a - air density in section a-a (equal on the adopted assumption ρ_0);

2) the mass flow rate, computed from the parameters of sections b-b and c-c according to equation (79):

$$m_c = \rho_c f \sqrt{u_b + \frac{2k}{k-1} \cdot \frac{p_b}{\rho_b}\left[1 - \left(\frac{p_b}{p_c}\right)^{\frac{k-1}{k}}\right]}, \tag{107}$$

where ρ_c and p_c are the assigned parameters of section c-c;

3) communication/connection between pressures p_a and p_b according to equation (101)

$$p_a^2 - p_b^2 = \rho_a u_a^2 p_a \left(\lambda \frac{L}{d} - 2 \ln \frac{p_b}{p_a}\right).$$

After substituting into the last/latter equation u_a from equation (106), we will obtain

$$p_a^2 - p_b^2 = \rho_a \frac{m^2}{\rho_a^2 f^2} p_a \left(\lambda \frac{L}{d} - 2 \ln \frac{p_b}{p_a}\right).$$

The last/latter equation together with equation (107) represents system of equations with two unknowns p_b and m, to solution of which is reduced our problem.

The approximate computations of the flow of gas in conduit/manifolds. Above it was shown that with sufficiently long conduit/manifolds the polytropic exponent on the strength of friction during the flow of gas was close (even in the case of complete heat insulation) to unity.

Page 277.

Assuming that the temperature will be retained constant, then constant it will be also the viscosity of air and, consequently, also Re whose value is necessary for the calculation of the coefficient of friction drag λ in expression (17).

Taking into account that which was indicated for the approximate computations of losses of head along the length of conduit/manifold can be applied the known formula of hydraulics (14) with the substitution of the average values of the entering it parameters. With use by this formula the gas conditionally is represented in the

form of the incompressible fluid, which has some average parameters:

$$\Delta p = p_1 - p_2 = \lambda \frac{L}{d} \cdot \frac{u^2}{2} \rho_{cp}, \qquad (108)$$

where p_1 and p_2 it are represented pressure in the end sections of the section in question; λ is the average for the cut of conduit/manifold in question coefficient of friction drag; $u = \frac{m}{f\rho_{cp}}$ - the average along the length of conduit/manifold air speed in conduit/manifold by sectional area f; $m = fu_{cp}\rho_{cp}$ - air mass flow; $\rho_{cp} = \frac{p_{cp}}{R_{cp}T_{cp}}$ - the average air density (in the first approximation, $\rho_{cp} = \frac{\rho_1 + \rho_2}{2}$); here R_{cp} is specific gas constant at mean pressure and temperature; ρ_1 and ρ_2 - density in the beginning and at the end of the section of conduit/manifold (approximately they accept $\rho_1 = \rho_2 = \rho$) in question.

The calculations show that the air flow in the channels of pneumatic system bears usually turbulent character (Re > 2300), in accordance with how coefficient λ they calculate according to the same expression, as during calculations of the channels of hydraulic systems.

In practice in 2300 < Re < 10^8 frequently is applied also the empirical formula, which considers the roughness of the surface of the conduit/manifold:

$$\lambda = 0,11\left(\frac{k}{d} + \frac{68}{\mathrm{Re}}\right)^{0,25},$$

where k/d is relative roughness of the internal surface of conduit/manifold; here k - absolute roughness; d - the inner diameter of conduit/manifold.

Value Re, entering this formula, is calculated in this case according to equation (11), which at the substitution of the mass flow rate

$$\dot{m} = uf\rho = \frac{uf}{v}$$

and of the average values of the entering parameters will take the form

$$\mathrm{Re}=\frac{4m}{\mu_{cp}\pi d}=\frac{4m}{\nu_{cp}\rho_{cp}\pi d},$$

where ρ_{cp} - the average density of the gas; d is a diameter of air duct; $\nu_{cp}=\frac{\mu_{cp}}{\rho_{cp}}$ and $\mu_{cp}=\nu_{cp}\rho_{cp}$ are the average values kinematic and dynamic viscosity.

Figure 225 gives the designed by the last/latter formula graph/diagrams of the dependence λ on Re for the conduit/manifolds of the different relative roughness k/d. Page 278.

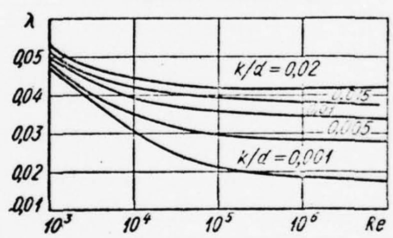

Fig. 225. Dependence of the drag coefficient on Re during air flow in conduit/manifolds with different relative roughness. ~~After accepting approximately~~

After accepting approximately

$$\rho_1 = \frac{p_1}{RT_1}; \quad \rho_2 = \frac{p_2}{RT_2}; \quad T_{cp} = \frac{T_1 + T_2}{2};$$

$$p_{cp} = \frac{p_1 + p_2}{2}; \quad \rho_{cp} = \frac{\rho_1 + \rho_2}{2} = \frac{p_1 + p_2}{2RT} = \frac{p_{cp}}{RT},$$

equation (108) can be presented in the form

$$p_1^2 - p_2^2 = \lambda \frac{L}{d} \cdot \frac{mRT_{cp}}{f^2},$$

where R is a gas constant at the average values of pressure p_{cp} and cf temperature $T_{cp} = \frac{T_1 + T_2}{2}$.

For the approximate computations of the flow of gas according to conduit/manifolds it is possible with sufficient accuracy/precision to use also the following simplified equation (under condition 0.9 <

$p_2/p_1 < 1$):

$$m = \mu f \sqrt{2\rho_1 (p_1 - p_2)}.$$

After substituting

$$\rho_1 = \frac{p_1}{RT_1},$$

we will obtain

$$m = \mu f \sqrt{\frac{2p_1}{RT_1} (p_1 - p_2)}. \quad (109)$$

Flow of the gas through the local friction. The local friction in the system of pneumatic actuator as in the system of hydraulic drive, play important role, since from the correctness of the estimation of the parameters of airflow, which takes place through the local friction, depend the accuracy/precision and the reliability of the calculations of pneumatic systems.

Unlike the flow of liquid, during which the energy of flow, expended for the overcoming of friction, is converted into thermal energy without its subsequent reverse/inverse conversion into mechanical energy, mechanical energy during the flow of air, being transformed into thermal, partially or completely is absorbed by flow itself (occurs the redistribution of energy). In view of this the mechanical energy of airflow, spent for the overcoming of friction, is not the forever lost energy, in other words, total energy of flow is retained constant/invariable, if we disregard heat exchange between the flow and the environment.

The calculation of the local friction of pneumatic system conducts in the general case according to equation (84). However, in view of the complexity of this equation frequently they use during the calculation of the gas flow through the local friction by

approximate equality (109).

For determining losses of head in the local friction they use in the general case known from hydraulics expression (19).

By using the law of flow continuity and by set/assuming the process of throttling/choking in the local friction polytropic, we will obtain as a result of conversions formula for air mass flow

$$m=\sqrt{\frac{2}{R}}\cdot\frac{1}{\sqrt{1+\zeta}}\,p_0\frac{1}{\sqrt{T_0}}\,f\sqrt{(1-x)\,x^{\frac{1}{n}}},$$

where p_0 and T_0 - pressure and the initial temperature of gas at the inlet into the local friction; f is a flow passage cross-sectional area of the local friction; x - the relation of air pressure.

Page 279.

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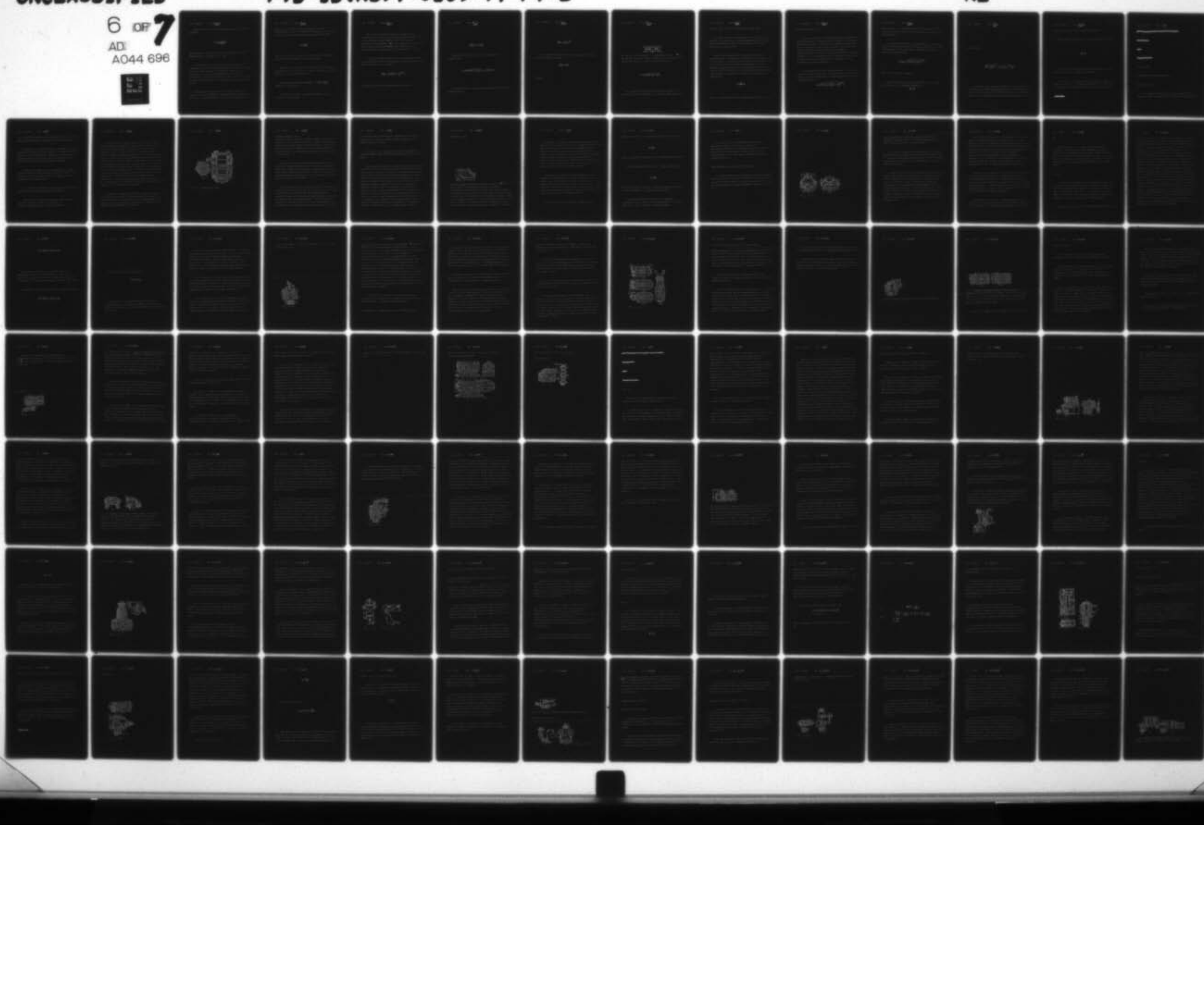

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Polytropic exponent can be determined, by considering expansion polytropic:

$$n=\frac{1}{1-\frac{\lg T_{\kappa}-\lg T_0}{\lg x}},$$

where T_{κ} is the final temperature of gas during expansion (it is determined from T-S-diagram for this gas).

Throttle/chokes. The mostd widely use local friction is the throttle/choke, with the aid of which changes the friction to passage of working medium (air) and is regulated the flow rate of this medium. With the aid of throttle/choke also are created the cell/elements of pneumatic automatics, in which conducts the addition of pressures, and also scaling the one pressure depending on another and etc.

Structurally the throttle/chokes of pneumatic systems are similar to the throttle/chokes of hydraulic systems. Specifically, are common the throttle/chokes in the form of opening/aperture in

washer (see Fig. 107a). Air mass flow through a similar throttle/choke is with $M < 1$ function of the ratio of the pressure of gas p_2/p_1 in the throttling cell/element (here m number Maxa):

$$m = f\left(\frac{p_2}{p_1}\right),$$

where p_1 and p_2 it are pressure before the throttling cell/element and after it (at output/yield from throttle/choke).

Accepting, that the process of the flow of the gas through this throttle/choke by section f (see Fig. 77a) the adiabatic, actual mass flow rate m of gas can be calculated:

a) during subcritical mode/conditions $1 > \left(\frac{p_2}{p_1}\right) > \left(\frac{p_2}{p_1}\right)_{\kappa p}$ - according to equation (83);

b) during supercritical mode/conditions ($p_2/p_1 < 0.528$) - according to equation (87).

Value μ can be calculated according to formulas for the incompressible fluid (see page 32). Virtually flow conditions of the gas through quadratic type throttle/chokes (see Fig. 77) is turbulent. So, accepting $Re_{\kappa p} = 2300$, it is possible to show that for washer throttle/chokes the turbulent mode/conditions with the possible pressure differentials will be observed with d > 0.2 mm.

As a result of certain complexity of the calculations according to equation (80) they use the approximate dependence, obtained on the basis of the fact that the function

$$f\left(\frac{p_2}{p_1}\right)=\frac{k}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}-\left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]$$

sufficiently accurately is approximated by the function

$$f_1\left(\frac{p_2}{p_1}\right)=\left(1-\frac{p_2}{p_1}\right)\frac{p_2}{p_1}.$$

Accordingly the mass flow rate of the gas through the throttle/choke

$$m=\mu f\sqrt{2p_1\rho_1}\sqrt{\left(1-\frac{p_2}{p_1}\right)\frac{p_2}{p_1}}=\mu f\sqrt{2\rho\,(p_1-p_2)}\,.$$

The critical ratio of the pressure of gas, with which occurs its maximum flow rate,

$$\left(\frac{p_2}{p_1}\right)_{\kappa p} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}.$$

For an adiabatic process the value of this expansion is equal, as this was shown on page 266,

$$\left(\frac{p_2}{p_1}\right)_{\kappa p} = 0{,}528.$$

Page 280.

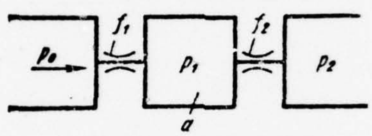

Fig. 226. The series connection of pneumos-throttle/choke. ¶ They use also the calculation formula, obtained by the transformation of formula (83) taking into account the isothermal process:

$$m = \mu f p_1 \sqrt{\frac{2}{RT} \cdot \frac{p_2}{p_1}\left(1 - \frac{p_2}{p_1}\right)}\,.$$

The described by the last/latter expression function of the expansion of gas has value $p_2/p_1 = 0.5$, which is sufficiently close

to value p_2/p_1 = 0.528, that corresponds to formula (85).

Are applied also the throttle/chokes, in which the flow rate is regulated by a change in the length of throttle channel. The mostd widely use construction of this throttle/choke is the helical throttle/choke (see Fig. 76a).

Similar throttle/chokes possess important for a practice positive quality - the stability of regulation, which consists in the fact that during the repeated installations of throttle/choke to one and the same position its discharge characteristic is retained virtually constant/invariable. In throttle/chokes of this type can occur with small pressure differentials laminar flow conditions whose characteristic is determined by the equation of Poiseuille [see equation (12)]

$$\dot{m} = \frac{\pi d^4 \rho}{128 \mu L} \Delta p,$$

where ρ - the air density, which usually accepts constant; μ - the

dynamic viscosity of air.

The series connection of throttle/chokes. By the consecutive and parallel connection of throttle/chokes are created the cell/elements of pneumatic automatics. In Fig. 226 are represented diagram of one of such cell/elements, that obtained the name of the pressure bag, which is the flowing chamber, formed by two consecutive throttle/chokes with an area of f_1 and f_2. This chamber possesses the property of the proportional reduction of pressures, thanks to which it enters as the basic cell/element in the schematics of many instruments of pneumatic automatics.

Let us assume that in input throttle/choke with an area of f_1 occurs the subcritical process of flow. In this case the flow rate through this throttle/choke will be determined according to equation [see equation (80)]

$$m = f_1 \sqrt{2\rho_0 p_0 \frac{k}{k-1}\left[\left(\frac{p_1}{p_0}\right)^{2/k} - \left(\frac{p_1}{p_0}\right)^{\frac{k+1}{k}}\right]},$$

where p_0 and ρ_0 - the assigned pressure and the density of the gas before the throttle/choke with an area of f_1; p_1 is pressure in the flowing chamber a.

Let us assume further that in output choke with an area of f_2 occurs the supercritical process of discharge. The flow rate through throttle/choke f_2 in this case will be determined according to equation [see equation (86)]

$$m = m_{max} = f_2 \sqrt{2\rho_1 p_1 \frac{k}{k+1}\left(\frac{2}{k+1}\right)^{\frac{2}{k-1}}},$$

where ρ_1 - gas density in chamber a.

Equalizing flow rates and raising both parts of the equality into square, and also taking into account that

$$\frac{p_0}{p_1} = \frac{\rho_0}{\rho_1},$$

we will obtain

$$\frac{\left(\frac{p_1}{p_0}\right)^{\frac{2}{k}}-\left(\frac{p_1}{p_0}\right)^{\frac{k+1}{k}}}{p_1/p_0}=\frac{k-1}{k+1}\left(\frac{2}{k+1}\right)^{\frac{2}{k-1}}\left(\frac{f_2}{f_1}\right)^2.$$

Consequently, during subcritical process in throttle/choke with an area of f_1 and during supercritical process in throttle/choke with an area of f_2 the ratio of the absolute pressure p_1 in the flowing chamber a to the absolute inlet pressure p_0 will be determined by the

relation of the areas of these throttle/chokes f_2/f_1.

During supercritical process in both throttle/chokes it occurs

$$\frac{p_1}{p_0}=\frac{f_1}{f_2}.$$

By analogy with electrical potentiometer by potentiometer the pressure bag is called pneumatic potentiometer.

The circuit of the examined choke cell/element is placed as the basis of a series of the automatic instruments of those possessing the properties of proportional reduction (see Fig. 244a and 253c).

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Pages 281-300.

CELL/ELEMENTS OF PNEUMATIC ACTUATORS.

Pneumatic engines.

On the principle of action and design the cell/elements of pneumatic systems are similar, with the exception of power supplies,

to the equivalent components of hydraulic systems, but frequently in both systems are applied the same types of these cell/elements.

Specifically, the energy conversion of the compressed air into mechanical work conducts in these systems by the volumetric pneumatic engines of rotary (compressed air motors) and rectilinear (power pneumatic cylinders) motion (Figs. 227 and 3). More rarely are applied pneumatic turners (moment pneumatic cylinders).

By volumetric pneumatic engine is understood the pneumatic engine, in which the energy conversion occurs in the process of the alternating filling of working chamber with working gas and his displacement from working chamber.

As the volumetric pneumatic engines (compressed air motors) of rotary motion are applied lamellar and gear machines.

Equipment/device and the calculations of compressors and pneumatic engines were examined in the course of "positive-displacement pumps and hydraulic engines".

Figure 227a gives the design concept of the standard lamellar engine (compressed air motor) of rotary motion. The compressed air will be fed through channel a of housing, and further through opening/apertures in stator 2 it enters the appropriate working chamber of motor, formed by two adjacent plates 3 and by the surfaces of stator 2 and of rotor 1, and, acting on these plates, develops the torsional moment. After the chamber of filling will be cut off during the rotation/revolution of rotor 1 from those which were connected with the window of the power supply of channels b in stator, filling by its compressed air ceases. During the further rotation/revolution of rotor the volume of the chamber increases ($q_1 < q_2$) and the being expanded air it continues to act on its restricting plates, developing the torsional moment. During the connection/compound of the chamber, filled by the partially enlarged air, with channels c of stator 2 air is driven out in the atmosphere.

The velocity of compressed air motor is regulated by means of the rotation of its stator 2, with which changes the duration of the connection/compound of working chambers with opening/apertures b of power supply and, consequently, also the degree of admission of the chambers by the compressed air. Page 282.

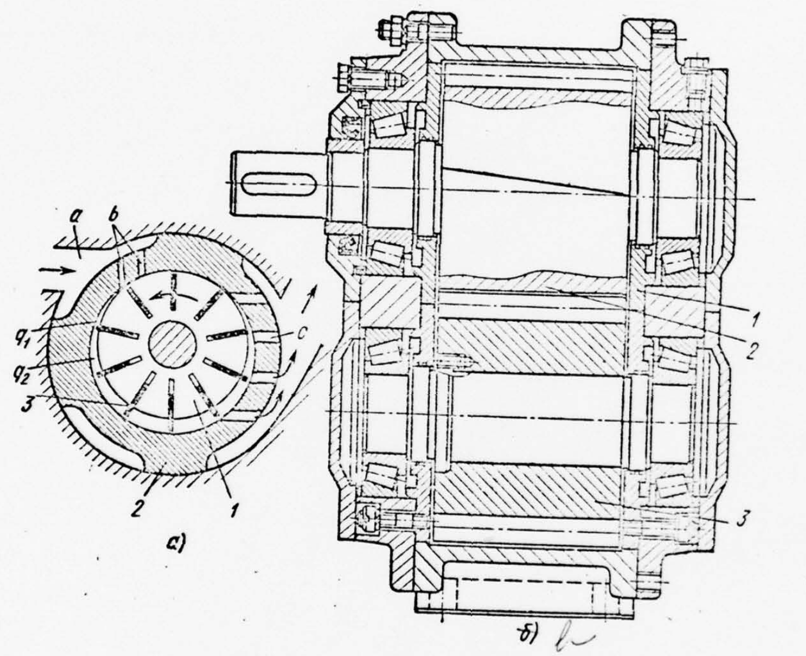


Fig. 227. Compressed air motors.

A similar pneumatic engine (pneumatic actuator), in which the control conducts by means of the cutoff of the flow of working gas, calls pneumatic engine (pneumatic actuator) with control cutoff, whereupon by cutoff is understood the cessation of the feed of working gas into the working chambers of pneumatic engine at the torque/moment, when still occurs an increase in their volume.

The work of compressed air motor can occur/flow/last, also, during the complete expansion of air up to a pressure of, close to atmospheric; however, in real machines is applied partial expansion, since complete expansion it conducts as a result of a considerable temperature decrease to an increase in the size/dimensions of machine and to the icing over of air ducts.

Figure 227b depicts the construction of gear type compressed air motor, which is the pair of helical gears 2 and 3 (angle of the slope of teeth 6-8°) whose cylinders are established/installed on antifriction bearings. For operational provisions under conditions of poor lubricant are applied lateral disks 1 of the antifriction material. The lubricant of gears in the work of compressed air motor is realized by the autolubricator, which feeds oil into the flow of

the compressed air, which through the tap/crane of control along channels in the housing of pneumatic engine is supplied to the block/module/unit of rotors.

In a series of the constructions of use of centrifugal type automatic devices, which restrict the maximum speed of the pneumatic motor.

Figure 228 shows the indicator diagram of the idealized process, in which the filling of cylinder occurs at the constant pressure p, equal main-line, and issue it occurs at the atmospheric pressure p_0. On the section of 1-2 ways of the displacement/movement of piston is filled by the compressed air; at point 2 power supply ceases, and begins the process of the expansion of air (curve 2-3 corresponds to expansion on adiabatic curve, and curve 2-3' - to expansion on isotherm); at point 3 (or 3') the cylinder is connected with the atmosphere, and the pressure instantly falls to the atmospheric p_0 (point 4), at which is displaced the air from cylinder; at point 5 cylinder is detached from the atmosphere, and the remaining in it air is compressed up to a pressure of p'; at point 6 cylinder again is connected with working main line, and pressure in it instantly is raised up to a pressure of p in the latter; further process is

repeated. Page 283.

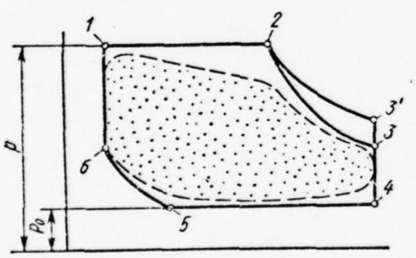


Fig. 228. Indicator diagram of compressed air motor.

Real process differs from described that which was idealized. Specifically, pressure curve of filling is not strictly parallel to the axle/axis of abscissas, but is sloped toward it (Fig. 228 depicts as broken line). Furthermore, on piston stroke it oscillates. Besides this, a pressure increase with filling of cylinder, and also decrease with issue occur not instantly, but during a certain time, which depends on different factors, including on the volume of dead space.

Fluctuations of pressure in cylinders are caused also by the imposition of the processes, which occur in adjacent cylinders. The latter caused by the fact that the air distributors usually are fulfilled so that with working main line simultaneously they are connected several cylinders, in one of which the process of filling approaches toward the end, and in other it only begins. It is obvious, because of this the air overflows from the cylinder with higher pressure in cylinder with smaller pressure, which can serve as a reason fluctuation of mean indicated pressure.

Accordingly, the idealized indicator diagram gives upper boundary of the effectiveness of pneumatic actuator (its maximally possible work) without taking into account of the losses of the compressed air. The relation of the area of actual indicator diagram (simplified form of this diagram is noted by point shading) to area idealized characterizes the quality of compressed air motor and is called the solidity ratio of indicator diagram.

The average value of the calculated torsional moment of

compressed air motor can be calculated according to equation (36):

$$M = \frac{N}{2\pi n},$$

where N is calculated rated power; n is rotation frequency in r/min.

After substituting value of $N = \Delta pQ = \Delta pqn$, we will obtain

$$M = \frac{\Delta pq}{2\pi},$$

where Δp - the average pressure differential, calculated according to indicator diagram; q is the working volume of motor.

The pressure differential can be for the approximate computations taken as equal to the height/altitude of the rectangle whose area is equal to the area of indicator diagram.

The velocity of the rotation/revolution of the rotor of compressed air motor is regulated by a change in the expenditure/consumption of the compressed air with the aid of the throttle/choke, included usually in input main line, and the torsional moment - by pressure change, realized by a regulator (reducer) of pressure.

Membrane/diaphragm actuating pneumos-mechanism.

In pneumatic systems, and in particular in pneumatic automatics with the small courses of actuating mechanisms and pressures (<10 kgf/cm²), are widely common the pneumoes-apparatus, based on the use of elastic cell/elements (diaphragm/membrane, bellows, etc.). Page 284.

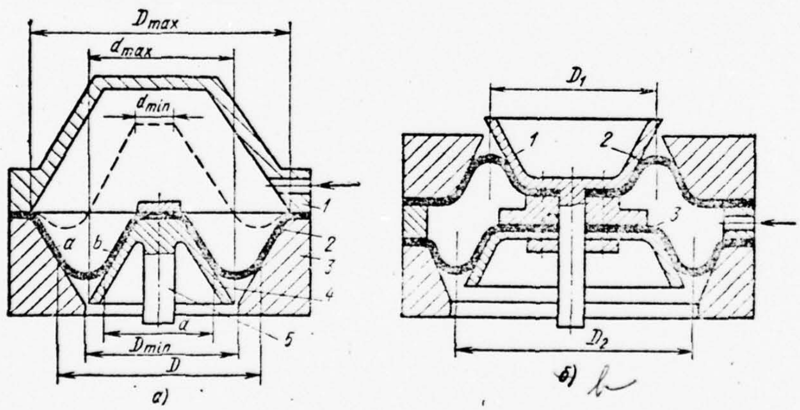


Fig. 229.

Fig. 229. Schematics of membrane/diaphragm node/units. These apparatuses are applied both as the actuating mechanisms and as sensing elements, receiving the measured value.

In the systems of industrial pneumatic automatics are applied predominantly the instruments with the elastic nonmetallic (rubber and plastic) diaphragm/membranes, which differ in terms of simplicity of construction, and also in terms of possibility of providing a complete hermetic seal.

Operating principle and basic calculations of these devices were examined above; however, in connection with work conditions under sensing elements of pneumatic automatics to membrane/diaphragm devices is presented a series of supplementary requirements. Specifically, the characteristics of the assemblies of these pneumos-instrument largely are defined by the properties of their elastic cell/elements - elastic diaphragm/membranes, defined both quality of material and by their properties to retain effective area under the varied conditions of work (values of course, drops of pressure, etc.).

The most important parameter is the constancy of the effective area of the diaphragm/membrane whose change is one of the basic sources of errors in the membrane/diaphragm mechanisms. These changes of the effective area in the function of a change in the jump/drop in pressure and displacement of the rigid center of diaphragm/membrane relative to the plane of its jamming are the fundamental characteristic of membrane/diaphragm pneumos-instrument. For providing a high sensitivity the diaphragm/membrane must have small flexural rigidity and be pliable with elongation in circular direction, but it must be sufficiently rigid with radial pull.

observance of the assigned law of a change in the effective area of membrane/diaphragm assembly. The examined above property of diaphragm/membranes to change its effective area in moving rigid center (see page 58), which for the majority of the cases of applying diaphragm/membranes is negative factor, since it restricts instrument accuracy, frequently is utilized for the creation of a series of the special instruments of pneumatic automatics.

Specifically, in the systems of pneumatic automatics it is required to ensure the smooth infinitely variable control (change) of the effective area of diaphragm/membrane according to the assigned

law.

Figure 229a depicts the schematic of membrane/diaphragm assembly, the effective area of diaphragm/membrane of which it is regulated by the displacement of rigid center. Membrane/diaphragm fabric 2 is jammed on the perimeter between the upper 1 and the lower 3 by parts of housing. The center section of the fabric is stopped up on mushroom-shaped stock/rod 4, which performs the role of rigid center.

Page 285.

The internal lateral surface of the lower part of housing 3 and the external lateral surface of rigid stock/rod 4 are executed along the assigned airfoil/profile (usually in the form of the truncated right cone with equal angles of taper at apex/vertex, by components 60°). The effort/force of the pressure applied air is transferred to output/yield through the stock/rod of 5 adjustable length.

Under the assumption that the section of free sagging on arc ab

(Fig. 229a) is outlined by circular arc and forming cones they are tangents to this arc, the effective area is determined by the instantaneous values of diameters D and d of the boundary lines of contact of the fabric of diaphragm/membrane with conical surfaces. The latter is caused by the fact that the sections of the membrane/diaphragm fabric, which lie on the conical surfaces of housing and rigid center, in the work of the membrane/diaphragm mechanism not of participants, whereupon with the equal angles of the conicity of the arcs of the curve of the section of sagging it is retained constant for all positions of rigid center along the axis. However, since the sagging of membrane/diaphragm fabric, which determines effective area of the membrane changes in moving rigid center in axial direction, changes with the position of this center also the effective area of diaphragm/membrane. Thus, for instance, during the maximum displacement of rigid center from certain current position, determined by the diameters of the contact of fabric D and d, upward (in Fig. 229a this position by represented broken line) the contact of diaphragm/membrane with the internal surface of the cone of housing 3 occurs according to diameters $D_{max} > D$ and with the external surface of the cone of rigid stock/rod 4 according to the diameter of the basis/base of cone $d_{max} > d$. Accordingly, the effective area of diaphragm/membrane reaches in this position of the maximum value, determined from the expression

$$S_{max} = \frac{\pi}{12}(D_{max}^2 + D_{max}d_{max} + d_{max}^2).$$

Analogously for the end lower position of rigid center instantaneous values d and D take values $d_{min} < d$ and $D_{min} < D$, whereupon d_{min} equal to the diameter of rigid center, a D_{min} - to the diameter of the lower opening/aperture of housing 3 (Fig. 229a).

Accordingly the minimum effective area of the diaphragm/membrane

$$S_{min} = \frac{\pi}{12}(D_{min}^2 + D_{min}d_{min} + d_{min}^2).$$

Range of a change in the effective area

$$\Delta S = S_{max} \div S_{min}.$$

Consequently, the minimum and maximum effective area of diaphragm/membrane and the range of a change in this area are determined by the smallest and greatest diameters of the basis/bases of these cones.

Analysis of equations and test results show that the effective area changes in moving rigid center from S_{min} to S_{max} over parabola (drawing of the fabric of diaphragm/membrane we disregard). Respectively the membrane tension, characterized by function S = f (x), where x - the displacement of center, is located for the schematic in question and the conditions linearly depending on the position of rigid center relative to its end lower position.

For the expansion of the range of a change in the effective area of membrane/diaphragm instrument are applied the schematics with two diaphragm/membranes, connected with common/general/total rigid center in the form of two truncated right cones equal to the conicity whose apex/vertexes are directed to each other (Fig. 222b).

It is not difficult to see that the effort/forces, developed with these diaphragm/membranes during the supplying of air into the chamber, are directed to opposite sides, in view of which the effective area of this bellows is equal to a difference in the effective areas of its diaphragm/membranes, which are determined as

in the earlier examined schematic, by the current effective diameters D_1 and D_2. Page 286.

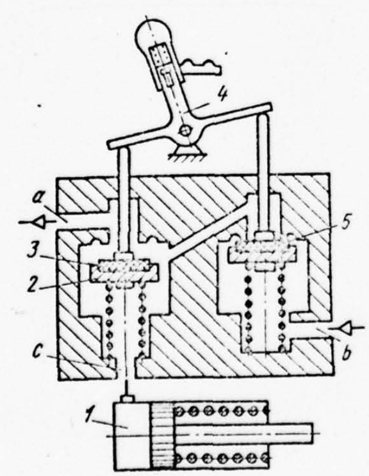


Fig. 230.

Fig. 230. Schematic of valve pneumatic distributor. Accordingly with the aid of this bellows can be obtained both the zero and negative effective area of diaphragm/membrane, i.e., the effort/force of this mechanism it can change its sign. In moving the common/general/total center of 1 diaphragm/membranes the down effective area of lower diaphragm/membrane 3 decreases, and upper 2 - increase. Since the effective area of an entire bellows in this schematic is equal to a difference in the effective areas of diaphragm/membranes 2 and 3, in certain position of rigid center 1, in which the areas of diaphragm/membranes are equal, the resulting effective area is equal to zero. During the further displacement of rigid center into the same direction the effective area of upper diaphragm/membrane 2 will become more, the lower 3 (effective area will become "negative"), in accordance with how will change the sign of effort/force on stock/rod.

Analysis shows that the effective area of this bellows in practice depends linearly on the displacement of its rigid center, and consequently, it possesses permanent hardness.

Distributive and regulating equipment for pneumatic systems.

In pneumatic systems is applied the distributive and controlling equipment for the same types and design concepts, as in hydraulic systems, and only sometimes are applied the supplementary means for an increase in airtightness and provision for a lubricant. Thus, for instance, everywhere are applied the distributive and other control valves with the supplementary means for sealing/pressurization in the form of rubber packing rings, and also valves with elastic shutters.

Distributing values. For air distribution are applied the on-off valves and valves with the flat/plane distributive cell/elements, which provide the high airtightness of connection/compound.

Figure 230 gives the schematic diagram of valve distributor for a control of the pneumatic cylinder of one-sided action. The supply of the compressed air conducts to channel b, and the outlet of mastered air - through channel a. The working cavity of actuating cylinder 1 is connected with channel c. During the rotation of lever 4 is meltted down one or another valve 2 or 5, as a result the air or will be fed into the working cavity of cylinder 1, of ii is abstract/removed from it in the atmosphere. The

sealing/pressurization of the gates of valves is realized by flat/plane or rubber teflon rings 3, sealed into the metallic parts of the gates.

The valve is controlled by hand by different levers and by pushers, and also by electromagnets and the compressed air, whereupon in the absence of the control pressure the gate of valve usually overlaps under the effect of spring the passage of air from that which supply to outlet.

Figure 231 depicts valves with the manual effect on pusher 2, realized through the lever by 1 (Fig. 231a), and mechanical effect on this pusher (Fig. 231c).

The compressed air (pressure to 6 kgf/cm²) will be fed to channel b (Fig. 231a). In the position of the gate of valve 4, shown in Fig. 231a (pusher 2 is not pressed), it by the effort/force of spring 5 and of the pressure of the compressed air is pressed against the saddle of bushing 6 and hermetically sealed overlaps with the aid of rubber packing 3 passage to air from inlet b to opening/aperture a, which is communicated through the axial boring of pusher 2 with the atmosphere. Page 287.

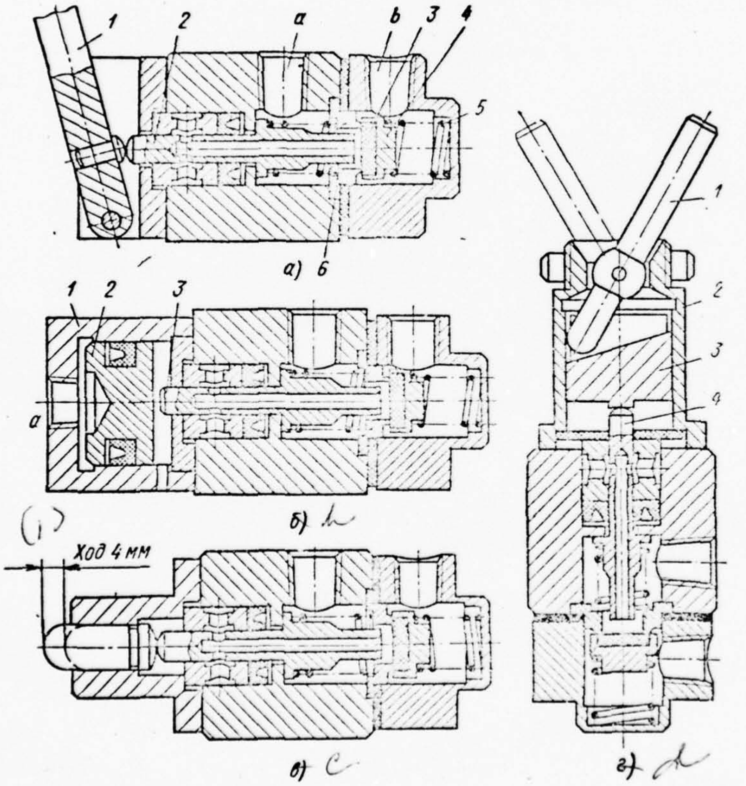


Fig. 231.

Fig. 231. The standard designs of air operated valves.

Key: (1). Course. With the submerging (displacement to the right) of pusher 2 it rests first by its end/face into rubber packing of 3 gates, disconnecting opening/aperture a from the atmosphere, and during further displacement it breaks away the gate of valve 4 from saddle, open/disclosing in this case passage to the compressed air from inlet b into opening/aperture a of pneumatic engine.

Figure 231b shows valve with pneumatic effect on pusher 3 through the piston of 2 pneumatic cylinders 1. The compressed air is supplied to channel a.

When it is required to ensure the fixing of gate in its both end positions, are applied different mechanical means. In the valve, presented in Fig. 231d, this is provided by the fact that lever 1 affects pusher 4 through that which is being moving in the guides 2 wobblers 3, with the aid of which lever 1 is record/fixed in both end its positions. This fixation is provided by the fact that in both end positions of lever 1 effort/force of the reaction of spring-opposed wobbler 3 attempts to retain/hold down it in this position.

Air operated valves with electromagnetic control. In the systems of pneumatic automatics widely are applieddistributive magnet valves.

Figure 232 gives the schematic diagram of pneumatic actuator with distribution by disk valves with rubber gates. Discovery/opening (melting down) valves 1 and 3 is realized by electromagnet 4 through lever 2 and coverage - by springs 7. Page 288.

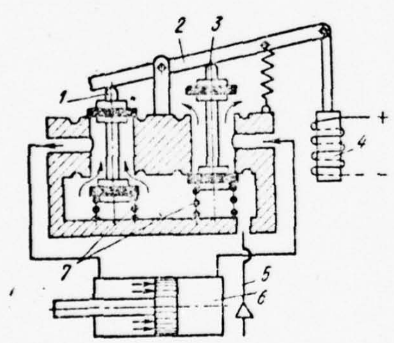


Fig. 232. Diagram of pneumatic actuator with valve distributor.

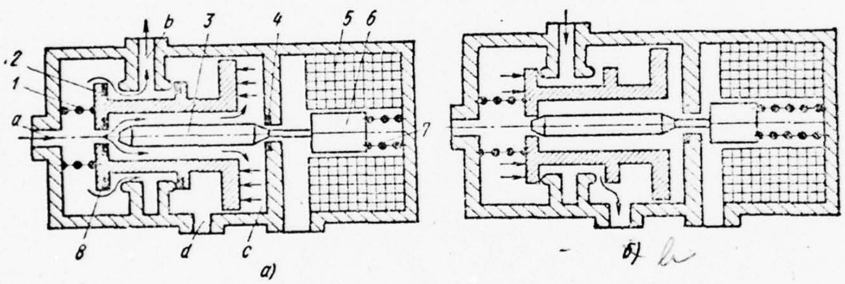


Fig. 233. Schematic of valve distributor with electromagnetic actuator. With melting down of left valve 1 and the release of right valve 3, the left cavity of pneumatic cylinder 6 is connected with the main line of 5 compressed air feed, but right it is connected with the atmosphere and vice versa.

Are applied the valves of direct action, also, with servo effect

(two-stage valves).

Figure 233a gives the schematic of two-position valve distributor (pneumatic stopcock) with electromagnetic servodrive.

The locking assembly of this distributor consists of two valves (gates), one of which 2, spring-loaded 1, is strictly the distributor, and the second 3, connected with the armature of 6 electromagnet 5, by pilot valve (pilot valve) by governing basic valve 2.

During the supplying of voltage/stress on magnet winding 5, its armature 6 will move, overcoming the effort/force of spring 7 to the right, as a result the right gate of pilot valve 3 will sit down on saddle 4, after going away from saddle 8, executed in basic valve 2 (Fig. 233a). In this case the air under pressure will enter chamber c and, acting on the right end/face of basic valve 2, it will move it to the left, after connecting the channel of power supply a with the channel of user b and after overlapping simultaneously passage to channel d, which drives in the atmosphere.

After de-energizing magnet windings 5 its armature 6 under spring effect 7 will move to the left, as a result the connected with it gate of pilot valve 3 will go away from saddle 4 and will connect chamber c with the atmosphere. This will allow spring 1 to return basic valve 2 to the initial (right) position (Fig. 233b). In this case valve 3 will sit down on saddle 8, overlapping air inlet from the channel of power supply a into chamber c.

Valves with flat/plane distributive cell/element. Are common pneumatic distributors with flat/plane distributive cell/element (valve), which are fulfilled with different (manual, pneumatic and electrical) control.

Structurally these valves are similar to the analogous valves of hydraulic systems.

Control by these distributors is realized as a rule, with the aid of pressure-operated devices or electromagnets.

Figure 234 shows fundamental pneumatic circuit with electropneumatic distributor in the form of slide valve 9 (see also Fig. ~~8~~ 54), given connected with it differential plunger 1 (diameter D > d). Page 289.

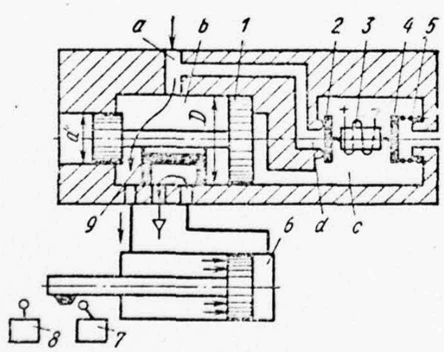


Fig. 234.

Fig. 234. Schematic diagram of pneumatic actuator with slide valve and electropneumatic control. ¶ Command cell/elements in this schematic are limit switches 7 and 8, which affect the detents, established/installed on the stock/rod of power pneumatic cylinder 6. Actuating cell/element is the magnet core 3, the carrier governing air operated valves 2 and 4, wrung out by spring 5 to the left position, in which valve 2 overlaps channel d, connected with the interpiston chamber b and the lead-in channel a.

During the retraction of core into coil air operated valve 4 detaches chamber c from the atmosphere, and valve 2 connects it with the channel of pneumatic system, as a result the compressed air, affecting differential plunger 1, moves slide valve 9 to the left position, in accordance with how the piston of pneumatic cylinder 6 is moved to the left.

At the end of the course of the stock/rod of pneumatic cylinder 6 by it is given into action disconnection switch 8, which de-energizes electromagnet 3, as a result valve 4 connects chamber c with the atmosphere, and valve 2 overlaps channel d, which connects it with chamber b between the differential pistons of listributive plunger 1. In this case the plunger together with valve 9 as a result

of action on its pistons of the unbalanced as a result of a difference in the areas pressure in the chamber is moved to the right (to the position, shown in Fig. 234), connecting the left cavity of pneumatic cylinder 6 with channel a of power supply. As a result the stock/rod of pneumatic cylinder 6 will return to the initial position, affecting at the end of its course disconnection switch 7. Further cycle will be repeated.

Figure 235a depicts the construction of valve with the bilateral pneumatic control of direct action.

Slide valve 5 is moved of one end position into another by pistons 1 and 4, common/general/total stock/rod of 2 of which is mechanically connected with valve. Against the mirror of the housing of 6 distributor valve 5 is pressed by the effort/force of spring 3 and by air pressure in chamber g, connected with channel c of power supply.

The compressed air will be fed to channel c and is abstract/removed in the atmosphere through channel e. The displacement of valve is realized by pressure of the compressed air,

applied alternately into the cylinders of bilateral piston from any source through opening/apertures a and b.

In one of the end positions of valve 5 compressed air it passes to opening/aperture f, connected with one of the cavities of pneumatic cylinder, and opening/aperture d at this time is communicated with the atmosphere. In the other end position of valve opening/aperture f is communicated with the atmosphere, and opening/aperture d of the second cavity of pneumatic actuator it is communicated with the main line of the compressed air. Consequently, opening/apertures f and d, connected with the cavities of power pneumatic cylinder, are communicated alternately in moving valve 5 either with the channel of power supply or with the atmosphere. It is necessary to consider that the valve in this schematic of distribution will be loaded with the air pressure, which acts on entire washed by it surface.

Figure 235b shows the construction of a two-position fourway air distributor of this type with bilateral electropneumatic control. The compressed air will be fed to the internal cavity c of the air distributor through the input opening/aperture h and further depending on the position of valve 5 proceeds to the appropriate

channels e or g, connected with the cavities pneumatic cylinder. Page 290.

Fig. 235. Constructions of slide valve with the control: a) pneumatic; b) electropneumatic.

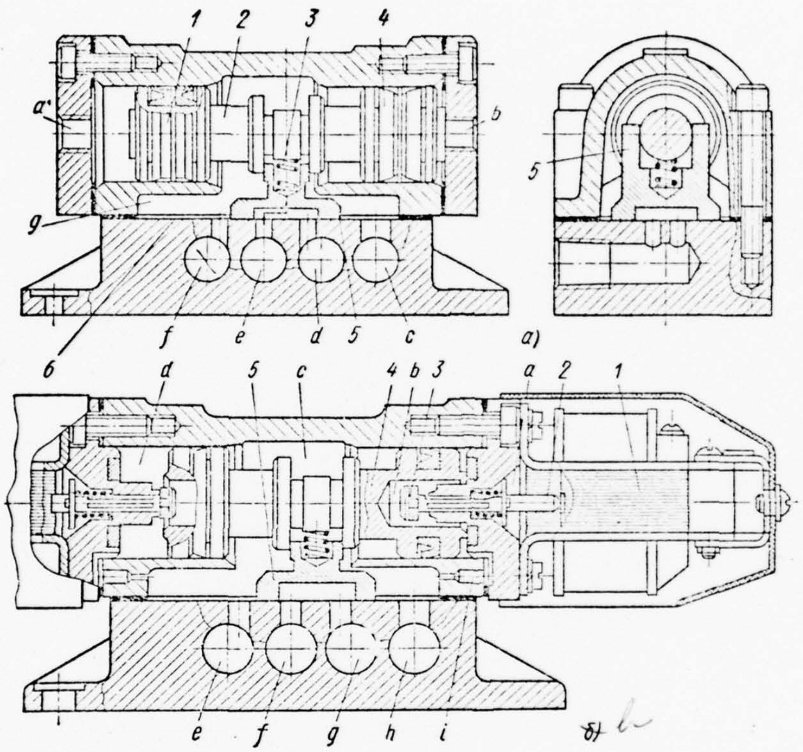
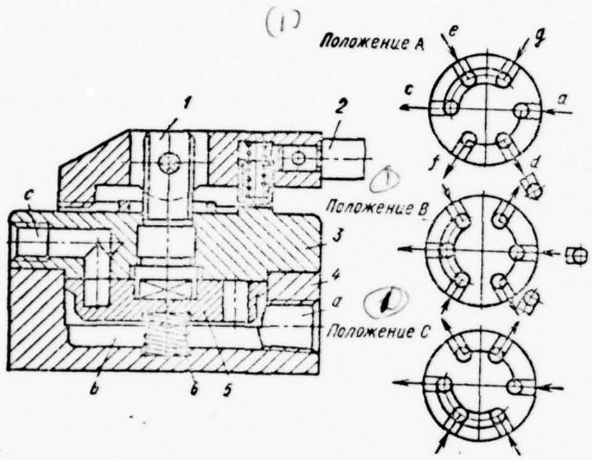


Fig. 236. Tap/crane of the series connection

Key: (1). Position



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Page 291.

Figure 235b right electromagnet 1 shows to the connected position, and left is shown in switched off.

The armature of the right electromagnet, being drawn in, presses on the rod of 2 control valves, open/disclosing gates. The compressed air from cavity c through opening/aperture i of a small sectional area (throttle/choke) enters cavity d and b under the end/faces of

piston 4. With the connected right electromagnet rod 3 together with rubber packing is moved aside to the left, open/disclosing output/yield to the compressed air from cavity b through opening/aperture a in the atmosphere. As a result the air pressure in cavity b falls, whereas in cavity d it equally to applied, in view of which piston 4 with valve 5 is moved to the right position. In this position of valve the compressed air from cavity c enters opening/aperture e, and opening/aperture g is communicated with the atmosphere through opening/aperture f. In this case the piston by 4 circular projections (snout) abut against rubber packing 6 and it closes output/yield to the compressed air through opening/aperture a in the atmosphere during entire time, when electromagnet it is included.

With the disconnection of the right electromagnet the rod (gate of valve) 3 returns to the initial position.

Upon the switching on of the left electromagnet cavity d is connected with the atmosphere and piston 4 together with valve 5 will move to the left. In this case the compressed air from cavity c enters opening/aperture g, and opening/aperture e is communicated with the atmosphere through opening/aperture f.

Tap/cranes of series connection. For the series connection of two power pneumatic cylinders during the motion of their stock/rods in one direction and for a simultaneous return in other direction, which frequently is required in the pneumatic actuators of machine tools and other machines, are applied multiposition rotary valves (tap/cranes) with flat/plane distributive cell/element. Distributor (Fig. 236) represents three-position six-running tap/crane with manual control. The compressed air will be fed to the tap/crane through channel a to the internal cavity b of tap/crane, formed by housing 4 and by cap/cover 3, and it is abstract/removed in the atmosphere through channel c. Against the mirror of cap/cover 3 is pressed by the pressure of the compressed air and by spring 6 slide valve 5 whose rotation conducts with the aid of cylinder 1 and crank 2, which has three fix/recorded positions, in which the channels of distributor d, e, f and g, which drive to the cavities of the controlled pneumatic cylinders (engines), they are communicated alternately with the cavity of pressure or with the atmosphere. In position of A inlet duct a is connected with channels d and f and outlet duct c with channels e and g; during the rotation of tap/crane counterclockwise (to position of B) inlet duct a is connected with channels d and g, but during the rotation of tap/crane clockwise

(position C) inlet duct is connected with channels g and e.

Regulating of the velocity of pneumatic engine and schematic of control. The regulation of the velocity of pneumatic engine is realized by the devices, used in hydraulic engines.

The velocity of operating unit usually is regulated by the choke speed governor, adjusted at the output/yield of engine (see Fig. 102b). During load variations in this case and, consequently, also the in speed of the displacement of output/yield changes the counterpressure in nonoperative (exhaust) cavity, which smooths speed fluctuation (raises the evenness of motion).

For maintaining the assigned mode/conditions are applied both the manual and automatic devices whose control is realized in the majority of cases in the function of pressure or way; more rarely is applied control in the function of time.

During the azimuth guidance of command/crew the function of pneumatic cylinder they enter from the control valves, which are

changed over in the appropriate place of the course moving cell/element of pneumatic cylinder or by another movable machine part. Page 292.

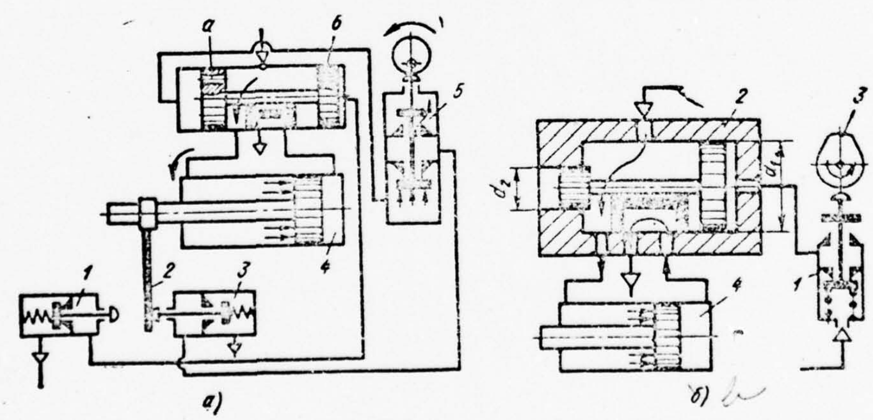

Fig. 237.

Fig. 237. Pneumatic actuators with the control: a) from way, b) from time.

The schematic of pneumatic actuator with this control is given in Fig. 237a. The cell/elements of pneumatic system are shown in the initial position, i.e., in the position, which precedes the beginning of cycle. System has starter valve 5 and two valves 1 and 3, controlled by detent 2 on the stock/rod of pneumatic cylinder 4, through which conducts the power supply of the servodrive of flat/plane distribution valve 6. The displacement of the latter is provided by pressure relief from the appropriate (right or left) cavity of the cylinders of servodrive. For this the compressed air constantly will be fed through the choke opening/apertures a in the small pistons of the plunger of valve 6 into both cavities of the cylinders (servodrive) of control of this valve.

System is put into action by hand with the aid of starter valve 5, in moving which the down left cavity of valve barrel 6 is connected through discovered valve 5 and through that which was opened by detent 2 on valve spindle 3 with the atmosphere, as a result the pressure in the left cavity of pneumatic cylinder 4 is lowered virtually to the atmospheric. Since valve 1 in this case remains enclosed, pressure in the right cavity of valve barrel 6 will be equal to pressure in grid/network, and the plunger of valve under the action of pressure differential will move to the left. In this

case the compressed air will be be fed into the left cavity of pneumatic cylinder 4, and mastered - driven out in the atmosphere. Under the action of the forces of pressure of the compressed air the piston of power pneumatic cylinder 4 will be moved to the right. In this case with the withdrawal of detent 2 from valve 3 latter under the action of internal spring will move to the left, after disconnecting the right cavity of driving/homing valve barrel 6 from feed line and after connecting this cavity with the atmosphere.

At the end of the foreward stroke of stock/rod detent 2 will give into action valve 1, open/disclosing output/yield to the compressed air from the right cavity of the cylinder of the administration of valve 6, as a result will occur its changeover to the position of the back stroke of the stock/rod of pneumatic cylinder 4. At the termination of this course the cycle will be repeated. The resistance of the choke opening/apertures a the small pistons of valve 6 must be largest possible (significantly higher the line resistance and valves 1 and 3).

Control in function of time. The control of pneumatic actuator in the function of time is characterized by the fact that the signals are supplied through the assigned in accordance with technological

process time intervals, which is reached with the aid of master switchhes and the devices (relay), which count off the duration of operations. Page 293.

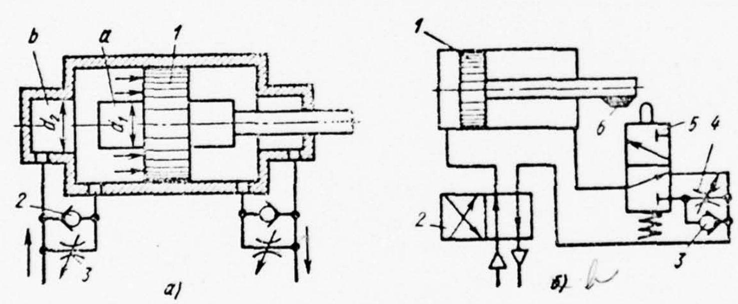

Fig. 238. Pneumatic actuators with braking device: a) internal; b) external. Figure 237b shows the schematic of the pneumatic actuator of the machine tocl, controlled with the aid of the established/installed on shaft master switch or another part of the machine tool, tumbler 3, the velocity of rotation/revolution of which determines the duration of cycle. With submerging by the

cam/catch/jaw of 3 valves 1 compressed air from grid/network enters the right cavity of distributor 2 with differential driving/homing piston ($d_1 > d_2$), moving it to the position, which corresponds to the working stroke of the stock/rod of pneumatic cylinder 4. During removal/taking control signal the plunger of distributor 2 with pressure of liquid in its left cavity returns to the initial (right) position.

Braking pneumatic actuators. In many instances of applying a pneumatic actuator it turns out to be necessary to carry out braking the piston of pneumatic cylinder at the end of its motion for the avoidance of the shock, which is undesirable or let us nonallow by the conditions of the strength of mechanisms or character of technological operation.

Braking is realized internal or by the external choke braking units, adjusted in the way of the air outlet from the cavity of emptying (nonoperative cavity). These devices decrease the flow area of exit (branch/drain) main line at certain point of the piston stroke of pneumatic cylinder, as a result in this cavity it is created the braking counterpressure, which retards of piston stroke.

Fig. 238 a shos the diagram of a pneumatic cylinder with a similar device. The feed of the cylinder and the movement of its piston takes place in the normal manner up until the corresponding projection a on piston 1 will not overlap one of the chambers b. After as piston of 1 cylinder will overlap during motion, for example to the left, by its projection a to chamber b, diameter d_2 of which is equal to diameter d_1 projection, air will be be able to driven out from exhaust cavity (in this case - right) only through adjustable throttle/choke 3 of small section, as a result the air in this cavity is compressed, braking motion with the intensity, determined by the control of throttle/choke.

To the initial (left) position the piston returns after the changeover of governing pneumatic distributor under the effect of the air, which takes place into the right cavity into the circuit/bypass of the throttle/choke through check valve 2. At the end of the piston stroke enters into action the braking system of the left cavity of cylinder. The kinetic energy of the driving masses of drive is converted during braking into the compression work of air. This work is determined by the amount of compressed in cavity cylinder of air and by the degree of its compression (by relation of counterpressure in cylinder at the end and beginning of braking), and also by the character of the process of compression (see page 260).

A deficiency/lack in internal braking devices is the difficulty of the control of the torque/moment of their switching on on piston stroke, in view of which if necessary for a similar control are applied external braking units. Figure 238b shows the schematic of drive with external braking unit. Page 294.

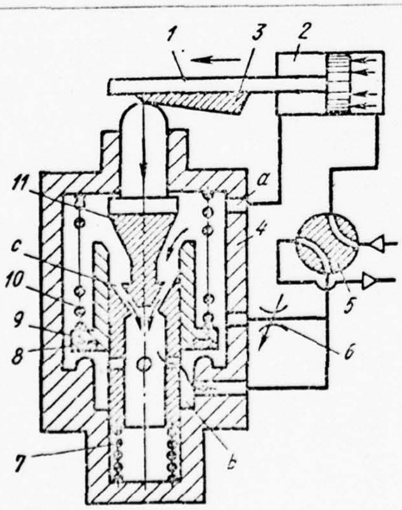

Fig. 239.

Fig. 239. Schematic of braking valve. System is equipped by fourway distributor by 2 and three-way distributors 5, controlled with the aid of cam/catch/jaw 6 on the piston rod of 1 pneumatic cylinder and by the equipped speed governor. In moving piston 1 to the right compressed air is abstract/removed through three-way 5 and fourway 2 distributors in the atmosphere. In the determined (assigned) place of course the piston rod with the aid of cam/catch/jaw 6 changes over (it melts down) distributor 5, as a result the air will emerge only through adjustable throttle/choke 4, in view of which is provided the deceleration of piston to the required value.

In the beginning of reverse/inverse piston stroke (to the left) the feed of the compressed air occurs before distributor 5 yet switched (are located in the embedded position), through distributor 2 and reverse/inverse dripped by 3, also, after the break-down of cam/catch/jaw 6 on distributor 5, indirectly avoiding throttle (velocity regulator 4. Distributor 5, reverse valve 3 and throttle 4 usually they are completed in one aggregate/unit, which obtained the name of braking valve. With the aid of this valve can be provided for both the required degree of throttling/choking at the end of the piston stroke and the change of the length of braking distance and speed of displacement of any section of way.

For braking the motion of the stock/rod of pneumatic cylinder by means of the throttling/choking of airflow, driven out from the emptyd cavity of cylinder, are applied also the special braking valves, which make it possible to inhibit the given operating unit (according to the assigned law) in any place of his way.

The schematic of a similar valve is shown in Fig. 239. Valve consists of throttle 11 with conical gate, controlled by detent (cam/catch/jaw) 3 on the motion work (stock/rod of 1 pneumatic cylinder 2) of machine, and freely sponsor check valve 9, spring-loaded 10. Plunger 11 in free state is held by spring 7 in the upper position, in which between the conical gate of plunger 11 and the case of valve 9 is formed the slot, on which the air from the being emptyd (nonoperative) cavity of pneumatic cylinder 2 is discarded through opening/apertures c, chamber b and distributing cock 5 in the atmosphere. In this case the stock/rod of 1 pneumatic cylinder 2 is moved at a velocity, determined by the initial position of plunger 11.

On the appropriate (assigned) section of way detent 3, connected

with the stock/rod of 1 pneumatic cylinder, presses on plunger 11 and, after pressing springs 7, it moves it down. In this case the plunger by 11 its conical parts goes in the opening/aperture of the shank of valve 9, as a result the flow passage cross-sectional area of slot d with the displacement of plunger 11 decreases, which causes an increase in the counterpressure in the emptyd cavity of pneumatic cylinder 2 and the smooth retardation of stock/rod 1. After plunger 11 completely will overlap passage slot, air from the emptied cavity of pneumatic cylinder 2 is displaced only through adjustable throttle/choke 6, by adjustment of which is regulated the speed of stock/rod 1.

During reversing distributing cock 5 compressed air from main line is supplied to cavity b. Page 295.

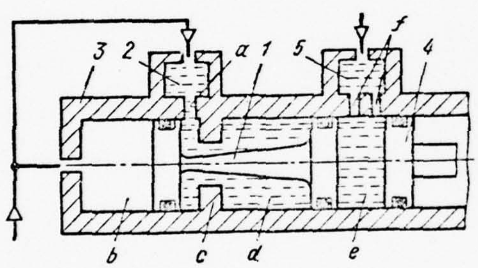

Fig. 240. Schematic of hydraulic damper. Overcoming the effort/force of weak spring 10, air it will discover valve 9 with packing layer 8 and will head into cavity, whence through opening/aperture a it will enter the left cavity of pneumatic cylinder 2. Thus is provided during reversing the unimpeded filling of pneumatic cylinder by the compressed air despite the fact that choke plunger 11 still remains for a certain period of time in the embedded position.

The evenness of braking and its duration are determined by length and the airfoil/profile of pressure/clamping cam/catch/jaw 3, which affects choke plunger 11.

Hydraulic dampers. As a result of compressibility of air to ensure with pneumatic damper the strict assigned law of braking and guarantee the cessation of piston accurately in end position is virtually impossible. In view of this in the case of the increased requirements for braking is applied hydraulic damper.

Figure 240 shows schematic of one of such dampers. The damper consists of a conical plunger 1 with two pistons, the interpiston chamber d of which is filled by brake juice. Piston by air pressure in the left cavity b, connected with air duct, constantly is held in end right position. Since with this same main line is connected small tank 2, under the same pressure will be located to the entrance of damper into action and the brake juice in chamber d.

After piston 4 of power pneumatic cylinder in moving to the left

will arrive to contact with the right brake piston and will set him in motion in the same direction, liquid from chamber d is displaced through the choke circular flow area, formed by stock/rod 1 and by opening/aperture in partition/baffle c, into the right cavity of brake cylinder 3 (into chamber a). Piston rod 1 usually is fulfilled in the form of the cone, with the selection of conicity of which it is possible to ensure the required effectiveness and the law of braking. The calculations show that with conical stock/rod 1 virtually is provided the law of the uniform deceleration of piston 4.

The reverse piston stroke of 4 pneumatic cylinders is realized through the liquid, which is located in small tank 5, into which for this will be fed through the air sparger under pressure.

The schematic in question prevents also the rigid shock of piston 4 with its approach with foreward stroke to the right piston of stock/rod 1. From schematic it follows that during piston stroke 4 to the left it displaces through opening/apertures f liquid from the intermediate chamber e into small tank 5. After the first of the opening/apertures f it will be overlapped by driving piston 4, damping effect will be raised, while after overlap and the second

opening/aperture piston 4 and stock/rod 1 will be connected by the liquid, closed in chamber e. Therefore device is provided the evenness of the start of damping device.

Pneumatic regulator. In machine-building wide application obtained the automatic pneumatic flow regulators, schematic of one of which was given in Fig. 241.

A change in the controlled parameter (expenditure/consumption of the air, which takes place on main line 14) is transferred in the form of signal (pressure change) to sensor 4, which moves the shutter/valve of 6 regulator, changing distance (clearance) x mejdu srezom sopla 5 and with shutter/valve. Page 296.

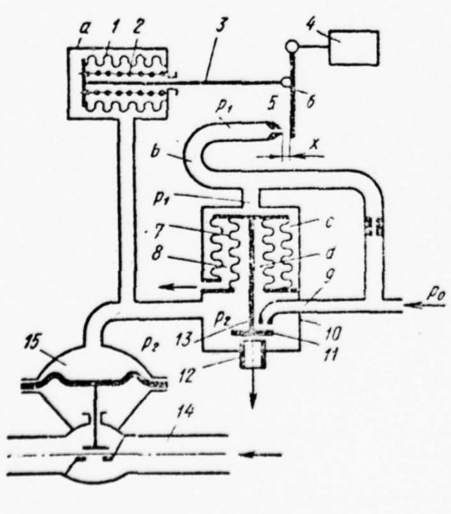

Fig. 241.

Fig. 241. Schematic of pneumatic regulator. As a result changes pressure p_1 in the pre-nozzle chamber b and in chamber c of bellows 8, whereupon during a decrease in distance x pressure p_1 is raised and vice versa. Accordingly changes also pressure p_2 at output/yield from regulator, which serves as the pressure of the medium, which feeds actuating mechanism (in this case the membrane/diaphragm drive of 15 overlapping valve/gate of main line 14). Pressure p_2 acts also on second bellows 7 whose area is less than the area of bellows 8.

For providing a state of equilibrium of process with the different values of controlled parameter in the schematic of regulator is included the feedback, which consists of chamber a with bellows 1 and spring 2. To chamber a is transferred pressure p_2 the adjustable (command) airflow, which affects bellows 1 and moves with the aid of stock/rod 3 shutter/valve 6, lowering the input signal of sensor 4.

In this schematic the nozzle - shutter/valve 5-6 is signal amplifier, which enters the bellows sensor a (amplifier of first stage), and bellows box 10 - by the amplifier of the second step/stage, which raises with the aid of adjustable throttle/choke 11-12 air pressure with p_1 to p_2 at the inlet into actuating

mechanism.

Let us examine the action of bellows box 10, which is the amplifier of the second step/stage. The compressed air under pressure p_0 through the adjustable throttle/choke of the type" nozzle" - shutter/valve 9-11 enters the bellows chamber d, connected on the one hand with the cavity of membrane/diaphragm drive 15 and simultaneously through the second adjustable throttle/choke, which consists of the same shutter/valve 11 and tube 12, with the atmosphere. Shutter/valve 11 by rod is connected with bellows box, in consequence of which its distance from the nozzle edge 9 and of the end/face of vent line 12 is determined by pressure p_1 amplifier of first stage (by pressure before nozzle 5). In moving shutter/valve 11 down increases the flow area between it and nozzle 9 and simultaneously decreases the section of the end-type slot between the shutter/valve and tube 12. During the complete overlap by the shutter/valve of 11 flow areas of tube 12 in chamber d and, consequently, also in membrane/diaphragm drive 15 will be establish/installed maximally the pressure.

The equilibrium of bellows sensor will be determined by the equality

$$p_1F_1 = p_2F_2,$$

where F_1 and F_2 - the effective areas of bellows 7 and 8. In view of the fact that $F_1 > F_2$, let us have $p_2 > p_1$.

By a regulator of this type is provided with a high degree of accuracy the linear dependence between a change in the controlled parameter and outlet pressure p_2, i.e., the displacement of the performing membrane/diaphragm of the drive of the overlapping valve/gate of main line 14. In other words, each value of controlled parameter corresponds the strictly defined value of pressure p_2 in membrane/diaphragm drive.

Apparatuses of the pressure adjustment of air. For the normal operation of pneumatic system its power supply must conduct by air with the constant pressure, which is provided by special pressure regulators (pressure reducers or reduction valves). Page 297.

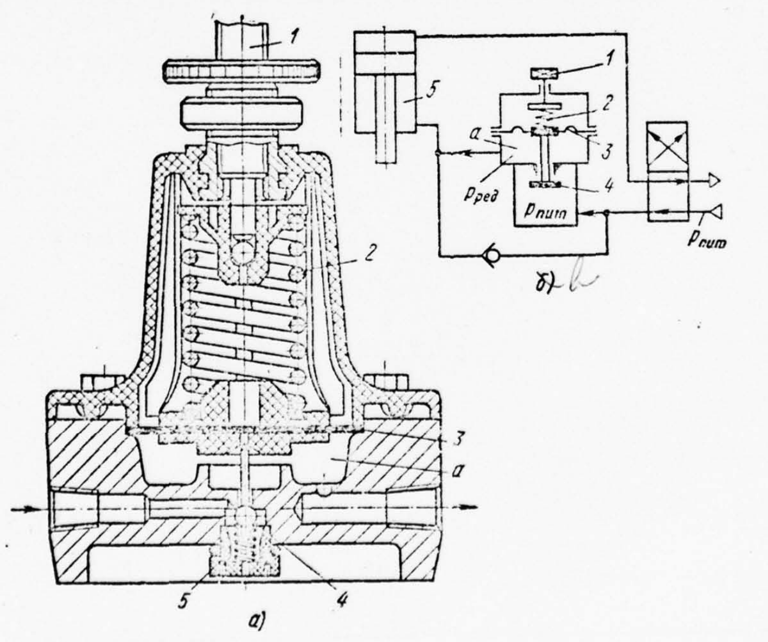

Fig. 242.

Fig. 242. Reducer (regulator) of air pressure (a) and the schematic of its application/use in pneumatic actuator (I). ¶ Pressure reducer is the regulator, employed for automatic decompression of the compressed air and his automatic maintaining at the assigned level.

As reducers they are applied predominantly their diaphragm (see Fig. 72b) and bellows (see Fig. 73) types. These reducers are utilized for obtaining relatively low pressures (to 30-40 kgf/cm²) as a result of the limited strength of diaphragm and bellows.

Gate in the valves, intended for a work with gases, is fulfilled usually in the form of the flat/plane (lamellar) rubberized or prepared from elastic material valve, seated to seat/socket with the rounded projecting edges (more rarely are applied ball bearing gates).

Figure 242a shows one of the apparatuses of this designation/purpose, which in practice it was called the name the stabilizer (reducer) of pressure (see also Fig. 72b). The principle of its work is based on automatic a change in the flow area of airflow during a change in pressure and air flow rate in power line

and maintaining, thus, the pressure constancy of air at the output/yield of stabilizer (in service line). The pressure constancy is provided by an automatic change in the position of the throttle valve, which controls the flow area of airflow during fluctuations of pressure in chamber a, connected with output/yield.

For the installation of the required operating pressure at the output/yield of stabilizer serves regulating (choke) screw/propeller 1, with the aid of which changes the effort/force of spring 2, which affects diaphragm/membrane 3, connected with valve 4, which is held into saddle by spring 5. A change in pressure and air flow rate in grid/network causes the displacement of diaphragm/membrane 3 and of valve 4, in consequence of which it changes the flow area of airflow until the forces, which affect diaphragm/membrane 3, are balanced and pressure in chamber a is not stabilized.

During a decrease in the pressure in chamber a, which can be caused by a decrease in the pressure in the supplying grid/network or by an increase in the air flow rate by users, diaphragm/membrane 3 under spring effect 2 will be drop/omitted and, after moving down valve 4, will increase the flow area of airflow, that will ensure pressure balance in chamber a to that which was assigned. Page 298.

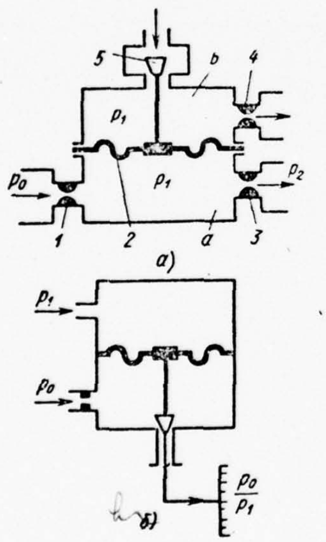

Fig. 243.

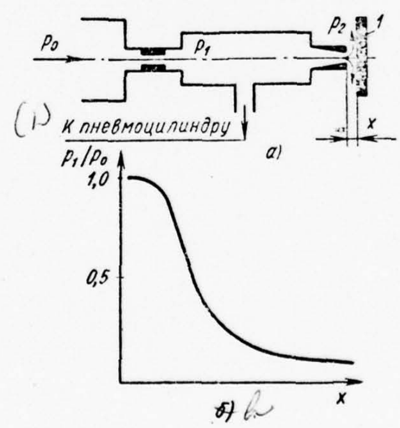

Fig. 244.

Fig. 243. Pneumos-instrument of pressure adjustment.

Fig. 244. Flowing pressure bag with a throttle/choke of the type" nozzle" - shutter/valve.

Key: (1). To pneumatic cylinder. ¶ An increase in the pressure in chamber a will cause the reverse/inverse action of the indicated parts of the stabilizer. A least change of the pressure in chamber a will cause an instantaneous change in the position of valve 4.

Figure 242b depicts the schematic of pneumatic system with the diaphragm regulator, established/installed in the main line of the power supply of the rod (lower) cavity of pneumatic cylinder 5, pressure in which must be lower than the pressure p_{num} in the main line of power supply $(p_{ped} < p_{num})$.

During a pressure increase p_{ped} air at output/yield from regulator (in chamber a) diaphragm/membrane 3 is deflected upward, and, overcoming the effort/force of spring 2 and moving the upward flat/plane gate of valve 4, decreases its flow area. With lowering in the pressure at output/yield, the flow area of valve increases. The

adjustment of regulator to the required pressure is realized by screw/propeller 1, compressive spring 2.

Proportional reduction of pressure. In the systems of pneumatic automatics are applied also the instruments of pressure adjustment, which possess the property of proportional reduction (reduction of pressure in the assigned relation), which is achieved by the application/use of series-connected two and more throttle/chokes in conjunction with reduction valve.

The schematic diagram of a similar regulator is given in Fig. 242a. Regulator consists of two chambers a and b, divided by diaphragm/membrane 2, connected with reduction valve 5, established/installed in upper chamber b. At the inlet into the lower flowing chamber a and at output/yield from it are established/installed the throttle/chokes of 1 and 3 fixed resistor. At output/yield from upper chamber is established/installed throttle/choke 4.

In lower chamber acts pressure p_1, determined by inequality $p_0 > p_1 > p_2$, where p_0 acts pressure before throttle/choke 1 and p_2 are

reduced pressure 3.

By the selection of the sections of these throttle/chokes f_1 and f_3 is establish/installed the required pressure p_1 in lower chamber a, which is supported also in the upper above-diaphragm chamber (we consider diaphragm/membrane absolutely elastic), by the selection of section f_4 throttle/choke at output/yield from upper chamber is regulated the air flow rate through reduction valve 5.

Page 299.

The calculation of the regulator in question is reduced on the calculation of filling and emptying of the chambers (tanks) a and b through throttle/chokes 1, 3, 4. Analysis shows that ratio p_1/p_0 pressures p_1 in chamber a and p_0 - at the inlet into this chamber will be determined by the relation of the areas of throttle/chokes f_3/f_1, whereupon during the supercritical process of the discharge in both throttle/chokes we have

$$\frac{p_1}{p_0}=\frac{f_1}{f_3}.$$

By analogy with electrical potentiometer the examined chamber usually they call pneumatic potentiometer.

On this same principle is constructed also the analogous sensor of the relation of pressure p_0/p_1 whose schematic is represented in Fig. 243b.

The flowing chamber with the exit adjustable throttle/choke of the type" nozzle", shutter/valve, widely is applied, in particular, in the pneudraulic servomechanisms as first stage of intensification (Fig. 244a; cm. also Fig. 127). Changing with the displacement of shutter/valve a distance x, i.e., changing the flow area of the exit

adjustable throttle/choke, we can control pressure p_1 in the flowing (interthrottle) chamber and in the connected with it pneumatic cylinder. Figure 244b shows experimental curve of function $p_1 = f\ (x)$ of this system.

At the low pressures of power supply ($p_0 \leq 0.5$ kgf/cm²) a pressure-operated device of this type can be approximately calculated as hydraulic, by utilizing an equation of the constancy of expenditure/consumption Q through both throttle/chokes.

In this case we have, according to equation (19),

$$p_0 - p_1 = \zeta_1 \frac{\rho}{2} \cdot \frac{Q^2}{f_1^2} \text{ и } p_1 - p_2 = \zeta_2 \frac{\rho}{2} \cdot \frac{Q^2}{f_2^2},$$

[и = and]

whence (set/assuming the drag coefficients of throttle/chokes $\zeta_1 = \zeta_2$)

$$\frac{p_1 - p_2}{p_0 - p_2} = \frac{\lambda}{1 + \left(\frac{f_2}{f_1}\right)^2},$$

or

$$\frac{\frac{p_1}{p_0} - \frac{p_2}{p_0}}{1 - \frac{p_2}{p_0}} = \frac{\lambda}{1 + \left(\frac{f_2}{f_1}\right)^2}; \quad \frac{p_1}{p_0} = \frac{p_2}{p_0} + \left(1 - \frac{p_2}{p_0}\right) \frac{1}{1 + \left(\frac{f_2}{f_1}\right)^2},$$

where

$$f_2 = \pi d_2 x;$$

$$f_1 = \frac{\pi d_1^2}{4},$$

here d_1 and d_2, diameters uncontrolled and adjusted of throttle/chokes,.

Electropneumatic relay and pressure indicator. For the checking of the pressure during pneumatic systems, realized by an effect on the contacts of the microswitch, connected in electrical control circuit, is applied the pressure relay. Relay is spring-loaded diaphragm/membrane 1, on which acts working air pressure (Fig. 245; see also Fig. 84c).

Air pressure, applied to channel a, acts through diaphragm/membrane 1 on cap 2 and pusher 5. If the effort/force, developed with air pressure, exceeds the effort/force of antagonistic spring 3 (effort/force of spring is regulated by screw/propeller 4), then pusher 5 is moved and affects the pin of microswitch 6.

The analogous in operating principle device, called pressure indicator, is applied for the feed of signal about the presence of pressure on the determined sections of pneumatic systems. Page 300.

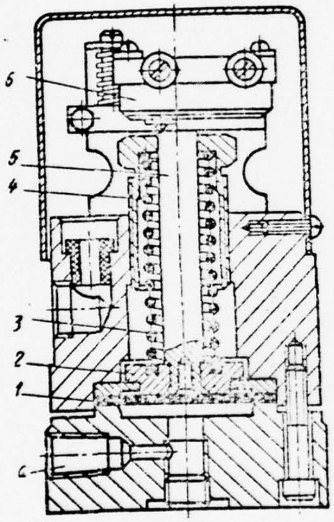

Fig. 245.

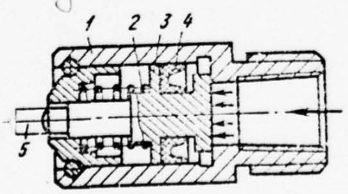

Fig. 246.

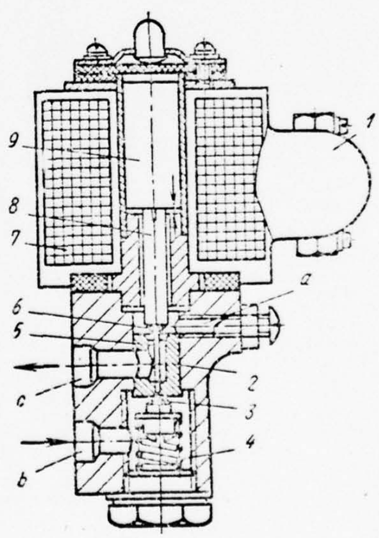

Fig. 247.

Fig. 245. Pressure relays.

Fig. 246. Pressure indicator.

Fig. 247. Electropneumatic valve/gate. This indicator is the miniature power pneumatic cylinder of one-sided action (Fig. 246). In the housing of 1 cylinder is moved small piston 3, packed by rubber sleeve 4.

In the absence of pressure on the controlled/inspected section of pneumatic system, small piston 3 under spring effect 2 is held in end right position. During the appearance of a pressure it, compressing spring 2, is moved to the left. Advanced stock/rod 5 is signaled with the aid of mechanical or electrical devices about the presence of pressure on that section of pneumo-circuit, to which is connected the pressure indicator.

Electropneumatic valve/gate. In the systems of pneumatic automatics are applied for a remote control of air ducts the valve/gates (Fig. 247), in which passage channels are closed and are

open/disclosed by the poppet valves, controlled by electromagnet.

Upon the connection/inclusion of the electric current, applied through terminal box 1, armature 9 steps down and with the aid of pusher 8 and valve 6 overlaps opening/aperture in the upper valve seat 2 and, simultaneously pressing through pusher 5 on foot valve 3, it open/discloses the lower opening/aperture of this saddle. In this case the compressed air entering from main line opening/aperture b, is headed through opening/aperture c toward user.

During de-energizing coil 7 spring 4 moves upward foot valve 3, overlapping air intake from main line. Simultaneously with this is moved upward overhead valve 6, providing jettisoning through the channel in valve seat 2 and tap a of air from the system of user in the atmosphere.

Pages 301-320.

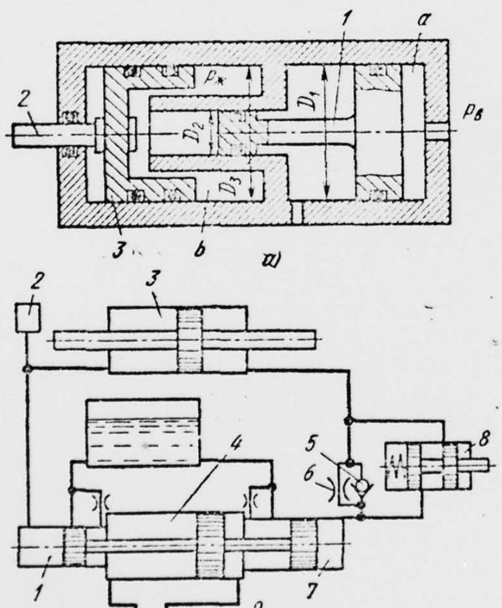

Fig. 248.

Fig. 248. Hydropneumatic pressure transducer (a) and the diagram of its application/use in pneumatic system (b). ¶ Hydropneumatic pressure transducers. In those cases when the available air pressure insufficiently for obtaining in the suitable size/dimensions of the pneumatic assemblies of the required effort/forces, apply the pneudraulic converters, in which relatively low air pressure is transformed into the high pressure of the liquid, which is the working medium of the hydraulic part of the system. Specifically, such amplifiers are applied in the feed units of metal-working machines, whereupon during their application/use it is represented possible to obtain the considerable effort/forces of feed at small air pressure.

Figure 248a shows the schematic of the widespread pneudraulic pressure transducer. The compressed air under pressure p_{B} is supplied to the left cavity a of the cylinder of converter and, acting on differential piston 1, forces out liquid under elevated pressure in chamber b. This pressure acts on piston 3, developing effort/force on stock/rod 2.

Pressure of liquid in chamber b

$$p_{ж} = p_{в} \frac{D_1^2}{D_2^2}.$$

Effort/force on stock/rod 2

$$P = p_{ж} \frac{\pi}{4} D_3^2 = \frac{\pi}{4} \cdot \frac{D_1^2 D_3^2}{D_2^2} p_{в}.$$

Figure 248b gives the schematic of the feed unit of drill press, which switches on pressure transducer, large-diameter 4 pneumatic cylinder of which sets in motion of two hydraulic cylinders of 1 and 7 small diameter, that feed by liquid the performing hydraulic power

cylinder of 3 feed units of machine tool.

Pressure p_2 in the cavities of hydraulic cylinders 1 and 7 is higher than the air pressure p_1, applied into pneumatic cylinder 4. It is proportional to relation to large F and by small f of the area of the differential piston:

$$p_2 = p_1 \frac{F}{f}.$$

The compressed air they will be distributed by slide valve 9 with electromagnetic actuator. During the supplying of air into the right cavity of pneumatic cylinder 4 its piston is moved to the left, as a result the piston of the hydraulic cylinder of 1 converter supplies liquid to the power of hydraulic cylinder 3 feed units of machine tool.

Simultaneously the liquid of their right cavity of hydraulic cylinder 3 is displaced through the speed governor, which consists of throttle/choke 6 and check valve 5, into the right cavity of the hydraulic cylinder of 7 converter.

Speed governor has adjustable throttle/choke 6, reverse/inverse valve 5 and bypass valve (bypass valve) 8 with electromagnetic control at the signals of sensor 2. In the beginning of the piston stroke of hydraulic power cylinder 3 liquid is displaced by it through the open in the initial position channel of valve 8 into the circuit/bypass of throttle 6 and after the switching on of the driving/homing electromagnet of valve and overlap of it by the plunger of drain channel - through throttle/choke 6.

The first part of the course is the uncontrolled in-rapid traverse of instrument to article and the second - by the adjustable working course. Page 302.

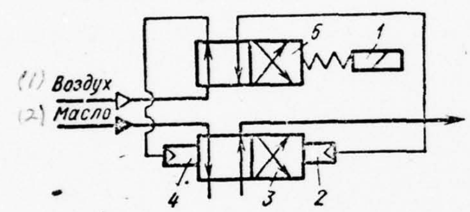

Fig. 249. Diagram of hydraulic valve with pneumatic control.

Key: (1). Air. (2). Oil.

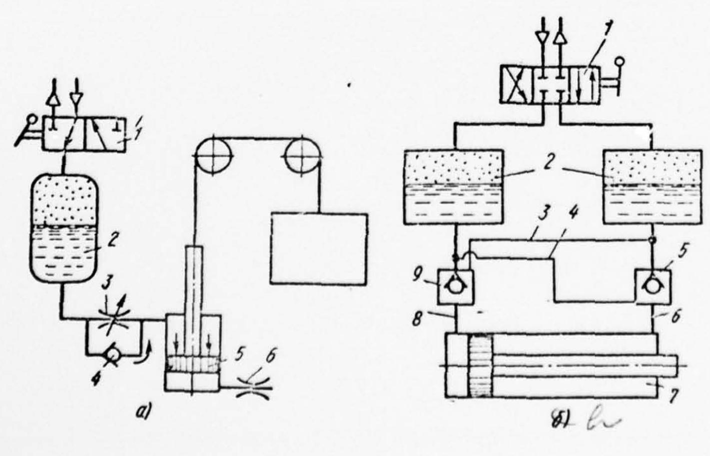

Fig. 250.

Fig. 250. Diagrams of pneumatic actuators with hydraulic retarders. With the reverse/inverse piston stroke of pneumatic cylinder 4 to the right the liquid from the hydraulic cylinder of 7 converter enters through check valve 5 into the right cavity of hydraulic cylinder 3, as a result the arbor of machine tool accelerated is moved to the initial position.

STANDARD PNEUMATIC ACTUATORS.

Combined pneudraulic drives.

Are applied also diagrams with combination pneumatic tires and hydraulics. The simplest case is the application/use of the compressed air for the drive of the distribution valve of hydraulic system.

Figure 249 gives the schematic diagram of a similar valve with the pneumatic first step/stage of intensification. The basic valve of 3 hydraulic systems is driven by two pneumatic servocylinders 4 and 2, controlled by pneumatic distributor 5 with electromagnet 1.

Depending on the position of magnet core 1, which sets to motion the auxiliary pneumatic slide valve, compressed the air is supplied to pneumatic cylinders 4 or 2, than and is realized the required control of basic distribution valve 3.

Pneumatic actuator with hydraulic retarder.

As a result of the high compressibility of air the control of the velocity of pneumatic servomotor, and in particular the provision for the assigned law of piston stroke, is extremely difficult. In view of this for the control of the velocity of pneumatic servomotors are applied hydraulic regulators. In the similar combined pneudraulic systems as energy source serves the compressed air, and the control of the velocity of piston stroke is provided with the aid of hydraulic devices.

Figure 250a depicts the diagram of this drive, intended for the uncontrolled accelerated and adjustable with the aid of

throttle/choke 3 depressions of the shutter/valve of metallurgical furnace. Page 303.

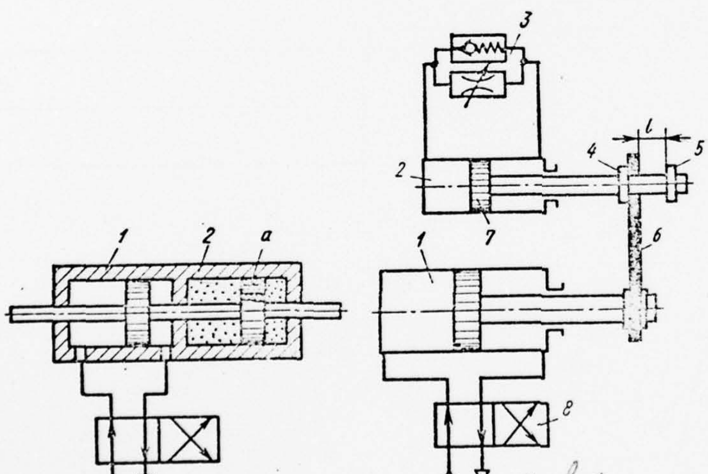

Fig. 251.

Fig. 251. Diagrams of pneumatic actuators with hydraulic brake. Upon the start of two-pass distribution valve 1 compressed air from power-supply system enters oleo-pneumatic reservoir (intermediary) 2 and, displacing from it liquid into the rod cavity of hydraulic cylinder, it moves piston down, heaving shutter/valve.

In order to remove the limitation of the rate of climb of shutter/valve, is established/installed in parallel with throttle/choke 3 check valve 4. In the nonoperative (lower) cavity of cylinder is established/installed air reactor 6, which creates backwater in this cavity, which facilitates the stability of motion of piston 5.

During the exchange of valve 1 to the second position the air from reservoir 2 is driven out in the atmosphere, and shutter/valve under its own weight is omitted, displacing into tank liquid from the upper cavity of the cylinder through adjustable throttle/choke 3, with the aid of which is regulated the velocity of the depression of shutter/valve.

Analogous drive with two oleo-pneumatic intermediaries 2, the hydraulic power cylinder of bilateral action and three-position valve 1 in air duct is shown in Fig. 250b. In the mid-position of valve 1 line of the feed of the compressed air into both intermediaries 2 are overlapped. In the left position of distributor the compressed air is supplied in left intermediary 2, whence it displaces the liquid through check valve 9 and the conduit/manifold into the left cavity of hydraulic cylinder 7 and simultaneously through the bypass conduit/manifold into its right (rod) cavity. In view of the difference of piston clearances it in this case is moved to the right, whereupon effective area is the sectional area of stock/rod (see Fig. 28).

During the exchange of valve into the left position the compressed air enters right intermediary 2 and displaces from it the liquid through check valve 5 and conduit/manifold 6 into the right (rod) cavity of hydraulic cylinder 7. Simultaneously with this liquid through bypass conduit/manifold 3 proceeds to the controlled check valve (hydraulic lock) 9 (see also Fig. 87b), which, after connecting the left cavity of hydraulic cylinder 7 with the left intermediary, makes it possible of liquid to move away from this cavity in intermediary.

Are applied diagrams with two cylinders, one of which is power and the second braking. Figure 251a shows the diagram of similar pneudraulic drive with power pneumatic 1 and braking by hydraulic cylinders 2 whose pistons are placed to common/general/total stock/rod. Braking the piston stroke of pneumatic cylinder in this diagram is realized by the choke opening/aperture a in the piston of braking hydraulic cylinder 2.

Figure 251b shows the schematic of a similar mechanism, used for the control of the feed rate of the instrument of drill press. With the piston of the power pneumatic cylinder of 1 this diagram, controlled by pneumatic fourway distributor 8, is connected into parts of the piston travel of braking hydraulic cylinder 2, served by damper. Page 304.

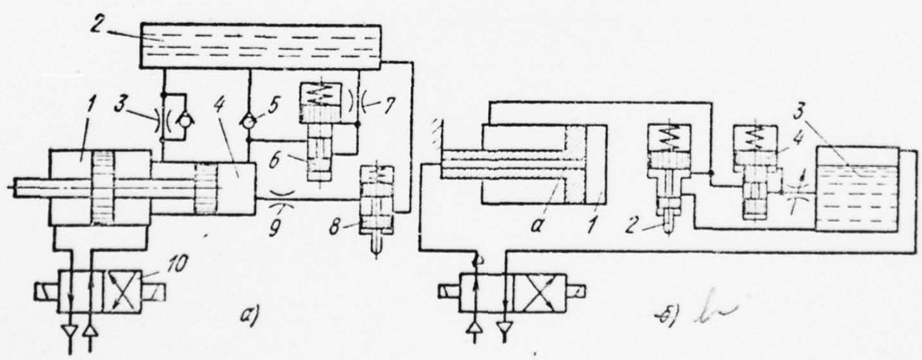

Fig. 252. Diagrams of pneumatic systems with the stabilizer of pressure. During the supplying of the compressed air into the

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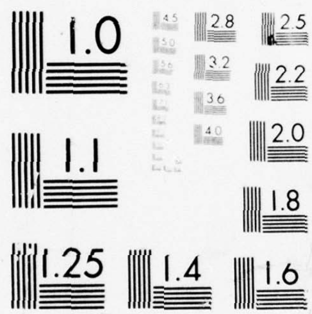

left cavity of pneumatic cylinder 1 its piston is moved to the right, whereupon in the first part of its course in the way l, limited by detents 4 and 5 on the stock/rod of hydraulic cylinder 2, it is moved accelerated, since the hydraulic damper this does not impede. With the arrival of lever 6, established/installed on the piston rod of pneumatic cylinder 1, at detent 5 velocity of the piston stroke of pneumatic cylinder 1 is limited by the friction of the throttle/choke of the speed governor 3, through which must be pressed the liquid from the left cavity of cylinder 2.

With the reverse/inverse piston stroke of pneumatic cylinder 1 its velocity is not limited, since the liquid, displaced from the left cavity of brake cylinder 2, passes through the return manuevering valve 3 into the circuit/bypass of its throttle/choke. To the arrival of lever 6 at detent 4 piston of 7 brake cylinder 2 is not moved.

For the control of velocity are applied also the stabilizers of pressure and the bypass valves, the first of which provide constant velocity with varying load and the second are a possibility of rapid adjustment with the subsequent slow shift. Figure 252a shows the schematic of the feed unit of drill press with the application/use of

the indicated device. In diagram is applied distributor 10 with electromagnetic control. During the supplying of the compressed air through this distributor into left through hydraulic cylinder 4 to the right. In this case the piston of hydraulic cylinder 4 displaces through reduction valve 6, throttle/chokes 7 and 9 and mechanically controlled bypass valve 8 oil from its right cavity into tank 2. The filling in this case with the liquid of the left cavity of hydraulic cylinder 4 occurs from tank 2 through the check valve.

During this period occurs the working shift of the piston of the pneumatic cylinder of 1 and connected with it feed unit of machine tool. The speed of this shift is determined by the setting up of throttle/chokes 7 and 9.

At certain assigned torque/moment bypass valve 8 with the aid of the cam/catch/jaw, established/installed on the feed unit of machine tool, is changed over to enclosed position, whereupon liquid is abstract/removed from hydraulic cylinder 4 into tank only through reduction valve 6 and arranged/located after it throttle/choke 7, as a result the piston stroke decelerates to the value, which corresponds to the control of this throttle/choke.

At the termination of working stroke distributor 10 is changed over, and the compressed air is supplied to the right cavity of pneumatic cylinder 1, moving its piston to the left.

Page 305.

Oil is displaced in this case of the left cavity of hydraulic cylinder 4 into tank 2 through throttle/choke 3, with the aid of which is regulated the velocity of back stroke.

Into the right cavity oil in this case enters through check valve 5.

The presence in the system of reduction valve 6 provides constant jump/drop in the pressure on throttle/choke 7 and, consequently, also constant fluid flow rate through it without depending on the load of pneumatic cylinder 1. This flow rate can be calculated for a throttle/choke in the form of opening/aperture in

fine/thin washer according to equation (20). Accepting also into consideration the expression $Q = v_\Gamma F_\Gamma$ which links flow rate of Q of the liquid through the throttle/choke with the velocity of motion v_Γ the piston of hydraulic cylinder and by its area F, we can write

$$v_z = \frac{Q}{F_z} = \frac{\mu f}{F_z}\sqrt{\frac{2\Delta p}{\rho}} = k\frac{f}{F_z},$$

where $k = \mu\sqrt{\frac{2\Delta p}{\rho}}$ — constant for these conditions value; here Δp is the pressure differential on throttle/choke 7.

During the establish/installed piston stroke effort/force on the stock/rod of pneumatic cylinder 1 without taking into account of friction

$$P = p_e F_e = \Delta p F_z + P_{нагр},$$

where p_B is pressure of the compressed air; F_B - the effective

piston clearance of pneumatic cylinder; Δp and F_{Γ} - a jump/drop in the pressure in hydraulic cylinder and the effective area of its piston; $P_{нагр}$ - the maximum force of feed (payload).

From this equation is determined the active area of the pneumatic cylinder

$$F_{\theta} = \frac{\Delta p F_{г} + P_{нагр}}{p_{\theta}}.$$

The minimum area F_{Γ} hydraulic cylinder 4 is selected taking into account the satisfactory conditions of the work of throttle/choke (see page 107). For this the minimum flow rate through the throttle/choke must be at the given speed of piston stroke

$$Q = v_{г} F_{г} \geqslant 5 \quad cm^3/s.$$

The diagram of the analogous pneudraulic feed system of the power pack of boring-boring machine is shown in Fig. 252b. In system is applied one cylinder 1, right cavity of which is pneumatic left - hydraulic, whereupon cylinder is fastened on the movable housing of the knob/cap of machine tool, and its stock/rod is connected with the stationary part of the machine tool.

For the shift of cylinder 1 to the right compressed air is supplied through the accomplished/carried out in stock/rod channel a to the right cavity of cylinder. Bypass valve 2 in the initial position of system is held by detent in the motion work of the machine tool in the embedded position, in which the liquid from the left cavity of cylinder 1 is displaced into tank without friction,

which corresponds to the rapid adjustment of the power pack of machine tool. After the break-down of detent on valve 2 liquid from the left cavity of cylinder 1 is displaced through the stabilizer of 4 velocities, as a result the velocity of cylinder descends to the assigned magnitude, determined by the control of the throttle/choke of this stabilizer.

The back stroke of power pack (course of cylinder to the left) is accomplished by means of the feed of the compressed air into tank 3, from which oil is extruded by the air through by-pass valve 2 into the left cavity of cylinder. Page 306.

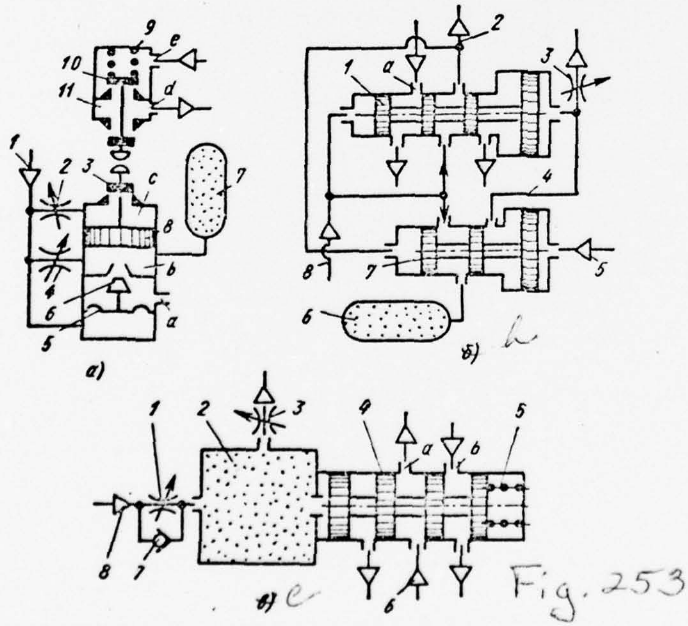

Fig. 253

Fig. 253. Diagrams of the timing relay. Instruments for providing the assigned cyclic recurrence of operations.

In many instances it is required to ensure in the operation of the system of any machine the assigned cyclic recurrence (assigned pauses) between courses. Specifically, in the discrete/digital control systems frequently it is required to ensure the assigned delay between the torque/moment of the feed (or removal) of signal and the beginning of the function of one or the other actuating element. The latter is reached with the aid of the timing relay, the operating principle of which is analogous to the action of the hydraulic relay of time (see Fig. 86). Structurally it differs from the hydraulic the fact that the sealing/pressurization is here executed by rubber blankets and collars. The duration (time) of holding is determined by time of the discharge of the certain volume through the adjustable throttle/choke.

For the creation of the pneumatic timing relays are utilized the processes of filling or emptying through the throttle/choke of certain volume (receiver). Time of increase or decompression to the assigned magnitude is regulated as in analogous hydraulic devices, by a change in the volume of receiver or flow area of throttle/choke.

Figure 253a gives diagram of one of the types of the timing relay, in which the provision for time is realized because of the filling of the receiver through the throttle/choke.

Relay consists of two throttle/chokes 2 and 4, receiver 7, command piston 8 and three-way valve distributor 11, through which is supplied performing pneumatic engine. The valve of 10 this devices is constantly enclosed and is put into action by the piston of 8 timing relay at the signals of air pressure in receiver 7. In the lower position of this piston channel d of pneumatic engine is overlapped.

During the launching/starting of system the compressed air enters through conduit/manifold 1, of which part of the flow is headed through adjustable throttle/choke 2 for the upper cavity c of cylinder and through adjustable throttle/choke 4 for the lower cavity b of this cylinder and receiver 7.

Page 307.

Simultaneously with this air enters under diaphragm/membrane 5 and, deflecting it, it intercept/detaches with the aid of valve 6 receiver 7 from the atmosphere (from channel a).

After certain time interval, determined by the volume of receiver 7 and by the control of throttle/choke 4, the pressure in receiver and, consequently, also in cavity b under piston 8 will be raised to the value, capable of overcoming the resisting forces, which act on piston 8. In this case the latter will move into the upper position and with its detent open/discloses valve 10. Simultaneously with this piston 8 open/discloses also valve 3, connecting cavity c with the atmosphere, thanks to which the further shift of this piston to upper end position will occur virtually instantly, and consequently, instantly will appear signal in the form of pressure in outlet duct d. After the removal of signal from input conduit/manifold 1 by communication of its with the atmosphere the pressure under diaphragm/membrane 5 will fall, as a result valve 6 will be discovered, connecting cavity b and receiver 7 with the atmosphere, as a result piston 8 under the spring effect of 9 distributor 11 will move into the lower position, in which channel d

is disconnected from channel e of power supply.

Figure 253b shows the diagram of the relay, the operating principle of which is based on the emptying of receiver. The compressed air proceeds to input conduit/manifold 8 it moves valve 1 to the right position, as a result conduit/manifold 2, which drives to the working cavity of performing pneumatic engine, is connected with delivery pipe 8 and channel a, connected with the second cavity of engine, it is connected with the atmosphere. The simultaneously compressed air, which enters from conduit/manifold 2 the left end/face of valve 7, moves him to the right position (represented in figure), whereupon begins filling of receiver 6 up to a pressure of main line.

During the supplying of signal in the form of the pressure of the compressed air to conduit/manifold 5 valve 7 as a result of a difference in the areas of right and the left of end/faces is changed over to the left position, as a result channel a is connected with main pressure line, but receiver 6 is through adjustable throttle/choke 3 with the atmosphere.

Through certain time interval the line pressure 4 is lowered as a result of the discharge of receiver to the value, with which the effort/force, which acts on the left end/face of valve 1, overcomes the resisting forces of device, and valve it is changed over to the right position, in which exit conduit/manifold 2 is connected with main line, and channel a it is connected with the atmosphere.

After the removal of control signal from input conduit/manifold 5 valve 7 is moved to the right position under the effect of pressure of the compressed air, supplied from conduit/manifold 2.

Receiver is connected with main line, and cycle is repeated.

Figure 253c depicts the diagram of the relay, in which the delay time is determined by the simultaneous control of filling and emptying of flowing receiver 2 of a constant volume. The compressed air from conduit/manifold 8 proceeds to adjustable throttle/choke 1, it fills receiver 2 and simultaneously through throttle/choke 3 emerges in the atmosphere.

During the appropriate control of the friction of throttle/chokes 1 and 3 (see also Fig. 266) pressure in receiver 2 will be raised, and upon achieving the value, sufficient for the overcoming of the resisting force of spring 5, valve 4 will be switched to the right position. In this position input main line 6 will be connected with channel b, and channel a it will be connected with the atmosphere. Since these channels are connected with the working cavities of performing pneumatic engine, this will cause a change in the direction of its motion.

After the removal of signals from conduit/manifold 8 receiver 2 rapidly is emptyd through check valve 7, as a result valve 4 under spring effect 5 returns to the left position, in which channel a is connected with main line, and channel b it is connected with the atmosphere.

With the impedance matching of throttle/chokes 1 and 3 and of the volume of receiver 2 it is possible to ensure in wide interval the preset time of holding. Page 308.

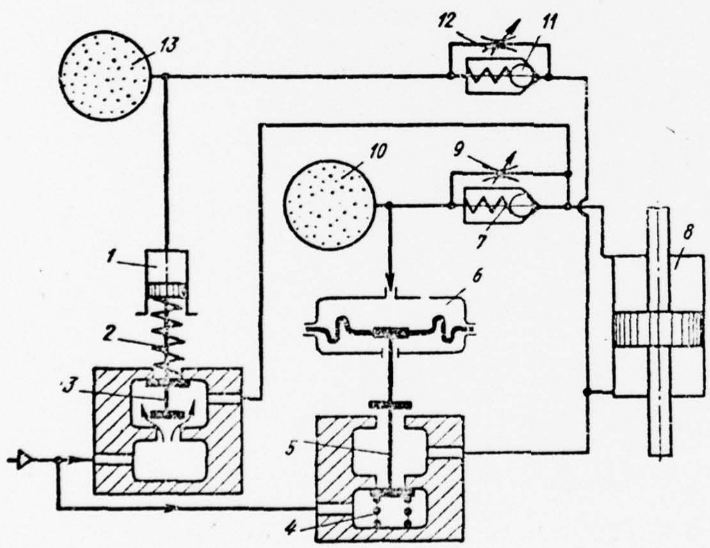

Fig. 254.

Fig. 254. Diagram of the pneumatic system, which ensures the assigned cyclic recurrence of operation. Experiment shows that during the sufficiently good decontamination of the compressed air the pneumatic timing relays are some of the most reliable and long-lived, but the practical stability of viscosity under different temperature conditions provides to them relatively high accuracy/precision and the stability of the duration of holding. The schematic of the device of a similar designation/purpose is represented in Fig. 254. In system are applied two auxiliary air receivers 10 and 13. Air comes from supply of power two by valve 3 and 5, one of which (3) is connected with the upper cavity of pneumatic cylinder 8 and the second with its lower cavity. Valve 3 by spring 2 is establish/installed in position (represented in Fig. 254), with which is open/disclosed the passage channel of the feed of the compressed air into the upper cavity of pneumatic cylinder 8. This channel overlaps with the aid of feed from auxiliary air receiver 13 into the cylinder of 1 air under signal pressure. In this case the lower channel of valve 3 overlaps, and upper, connected with the atmosphere, is open/disclosed. Second valve 5 connects the chamber of valve (upper window) with the atmosphere and overlaps the feeding channel, open/disclosing it with membrane/diaphragm drive 6.

From diagram it follows that at the initial moment of the action

of system the compressed air enters the upper cavity of cylinder 8, moving its piston down. The lower cavity of cylinder 8 at this time is connected through the upper window of valve 5 with the atmosphere. Simultaneously with this the compressed air enters through check valve 7 in in auxiliary receiver 10 and membrane/diaphragm drive 6. The volume of receiver 10 is selected by such, that a pressure increase in it to the value, necessary for the displacement of the diaphragm/membrane of drive 6, will occur for the interval of time, the equal to time of displacement/movement of the piston of actuating cylinder 8. As a result at the torque/moment of the termination of piston stroke valve 5 will be switched and the compressed air will head into the lower cavity of cylinder. Since to piston in this case acts the air pressure from both sides, it is found in rest that it corresponds to the first pause in work.

Simultaneously with air supply into the lower cavity of cylinder 8 will be begun the filling through the check valve of 11 supplementary receivers 13.

[redacted]

Page 309.

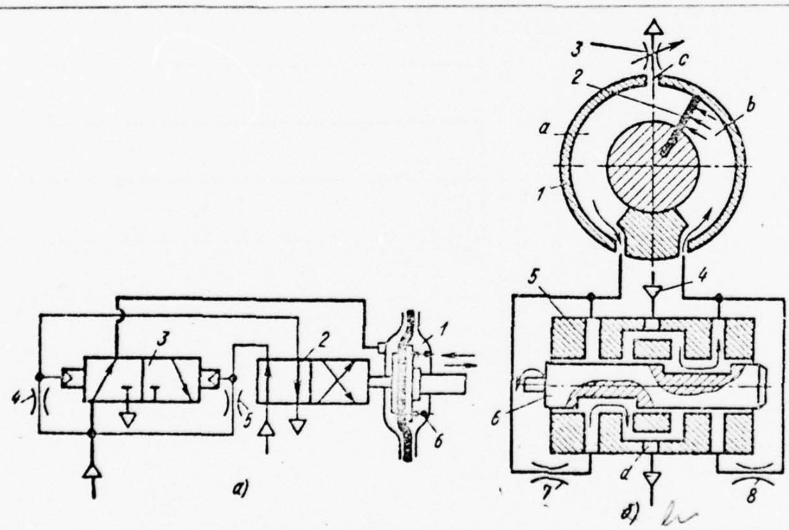

Fig. 255.

Fig. 255. Diagrams of the pneumatic actuators, which ensure the oscillatory motions of actuating mechanism. As soon as pressure in reservoir will become sufficient for the overcoming of the effort/force of the spring of 2 valves 3, the latter will be switched and will connect the upper cavity of cylinder 8 with the atmosphere, as a result piston it will begin to be moved upward.

Simultaneously with this will be begun slow overflowing through controlling throttle/choke 9 in the atmosphere of air from auxiliary receiver 10, as a result pressure in it and in membrane/diaphragm pneumatic actuator 6 will be lowered so, that valve 5 under spring effect 4 will be switched to the initial (opened) position, after connecting with the atmosphere and the lower cavity of cylinder 8. Since on both sides of the piston of this cylinder will be establish/installed the atmospheric pressure, it will be found in rest which corresponds to the second pause in the operation of system.

However, during the changeover of valve 3 with the atmosphere simultaneously is connected through adjustable throttle/choke 12 receiver 13, as a result of a pressure drop in which valve 3 will be switched to the initial (enclosed) position, by which the compressed

air again will enter the upper cavity of cylinder 8, which corresponds to the beginning of the repetition of cycle (after the holding of the assigned pause at the end of the course).

With the appropriate selection (calculation) of the volumes of supplementary receivers 10 and 13, and also of the friction of throttle/chokes 9 and 12 and of the effort/forces of springs 2 and 4 it is possible to ensure the assigned pauses in the piston stroke of cylinder 8.

Pneumatic actuators of continuous oscillatory motion.

For the automation of technological processes frequently are required the drives of continuous oscillatory or repeated motion. Specifically, the mechanisms of oscillatory motion are applied for the drive of bunker loaders and superfinishing tool heads on machine tools and etc.

Figure 255a gives the schematic diagram of mechanism for obtaining oscillatory motion from pneumatic engine is connected with

distributor 2, the governing three-way distributor 3 way of pressure relief. During the supplying of air from distributor 3 into pneumatic engine 1 its diaphragm, overcoming the effort/force of guard 6, is deflected and through the stock/rod changes over distributor 2, which through distributor 3 connects the diaphragm chamber with the atmosphere. As a result spring 6 deflects diaphragm to opposite side, moving distributor 2, which they will switch distributor 3 into the position of the power supply of pneumatic engine, and cycle it is repeated. Page 310.

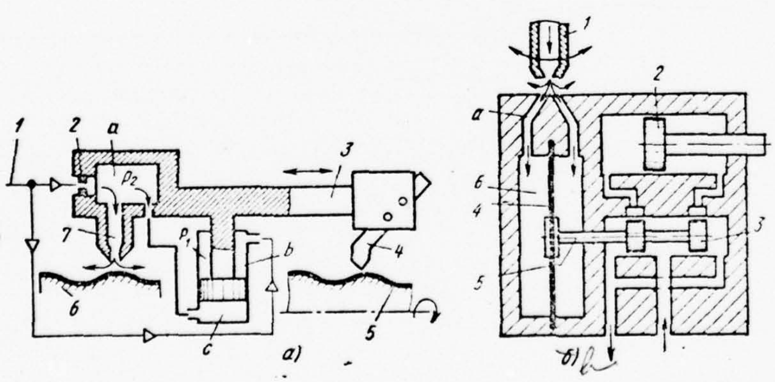

Fig. 256. Circuits of the follow-up amplifier: a) pneumatic; b) pneudraulic. Because of the stiffening joint of distributor 2 with

the stock/rod of diaphragm is provided the oscillatory motion whose frequency is regulated by throttle/chokes 5 and 4, established/installed on the inlet into the cavity of the administration of distributor 3.

Figure 255b depicts the schematic of pneumo-mechanism with the circular vibrations of the output/yield, used for the drive, for example, of the knob/cap of grinding machine. A mechanism has to distribute 5 with rotary slide valve and the pneumatic cylinder of 1 rotary action, working plate (blade) of 2 of which is connected through the output shaft with load. During the rotation/revolution of the valve of 6 distributor of chamber a and b of pneumatic cylinder consecutively they are connected through the channels of distributor with supply main 4 and with channel d, which drive in the atmosphere. The oscillation/vibration frequency is determined by the number of revolutions of valve 6, whereupon during the appropriate execution of the latter in its each of revolutions can occur two and more the vibrations of plate 2 and, consequently, also output shaft. With large number of revolutions the valve is establish/installed on needle bearing (see Fig. 54c).

The swing of the output shaft of engine is regulated by air

pressure, applied to distributor 5. Experiment shows that during a change of the air pressure in the range 0.1-4 kgf/cm² peak-to-peak of the oscillation of plate varies within the limits of 10-40° in work 15-50 Hz frequency.

In engine block on the axis of its symmetry is a opening/aperture with adjustable throttle/choke 3, which connects during the fluctation of the plate of chamber a and b with the atmosphere, thanks to which it is stabilized the initial value of the angle between the axis of the symmetry of housing and the axle/axis, relative to which oscillates plate. By the control of the friction of throttle/choke 3 it is possible to carry out a change within limits ± (5-7) ° the position of the axis of symmetry, relative to which oscillates the plate of engine.

Throttle/chokes 7 and 8 serve for braking plate 2.

pneumatic servomechanisms.

The compressed air is applied also as working medium and in

slave/servo type amplifier systems.

Figure 256a shows the schematic of the servomechanism of duplicating lathe with a distributor of the type the" nozzle" - shutter/valve whose nozzle of 7 is placed on the exit component/link of system. Shutter/valve in this diagram is the very template/pattern of 6 article 5, which is duplicated by cutting tool with high accuracy/precision. The compressed air from the main line of power supply 1 constantly will be fed into rod of cavity b of pneumatic cylinder, cavity c, opposite to stock/rod (its effective area usually 2 times exceeds the area of rod cavity), is connected with the main line of the power supply through throttle/choke 2, established/installed before nozzle 7.

Page 311.

Since in the channel, along which will be fed the compressed air into the nozzle chamber a, is established/installed throttle/choke 2, pressure p_2 in this chamber and, consequently, also in cavity c of pneumatic cylinder it changes during a change in the slot between the nozzle and the template/pattern. Since in the rod cavity b of

cylinder 7 pressure is constant (equal to pressure p_1 power supply), jump/drop $\Delta p = p_1 - p_2$ in cavities b and c of pneumatic cylinder changes with the change in the size/dimension of this slot, caused motion of instrument/tool carriage 3 along template/pattern, as a result of which carriage and, consequently, also cutting tool 4 repeat (they duplicate) during their motion the airfoil/profile of template/pattern 6. Thus, for instance, with an increase in the slot between the template/pattern and the nozzle the air flow rate from chamber a through this slot will exceed its admission into the chamber through throttle/choke 2, in consequence of which pressure p_2 in chamber a and in cavity c of pneumatic cylinder will fall, and piston with instrument and nozzle under effect of pressure p_1 in rod cavity will move to template/pattern (to the side of a decrease in the slot). During a decrease in this slot the piston moves to opposite side (from template/pattern).

Thus, nozzle and, consequently, also cutting tool it will "follow" with certain accuracy/precision the airfoil/profile of template/pattern, whereupon with constant load at the output/yield of pneumatic cylinder nozzle edge will be located on this distance from the surface of the template/pattern, with which the air flow rate through the formed slot between section/shear and template/pattern will be equal under all other identical conditions to the flow rate

through throttle/choke 2. Virtually this distance does not exceed several micrometers, whereupon since the viscosity of air during the possible fluctuations of temperature is retained virtually constant, the system provides under the static conditions the high accuracy/precision of tracking.

The accuracy/precision and the sensitivity of tracking are determined in the diagram in question by the same factors, as in hydraulic drive, and also by the compressibility of the working medium (air), which increases delay in final adjustment by the output/yield of the signals of the inlet.

Are applied slave/servo type also combined pneudraulic amplifiers, in which first stage of intensification is pneumatic, and the second - hydraulic. Figure 256b gives the circuit of a similar two-stage amplifier, used in automatic control system of aircraft in air.

First stage of pneumatic type intensification consists of the jet pipe of membrane/diaphragm type 1 and pneumatic actuating element with the receiving windows a (see also Fig. 152a). Diaphragm/membrane

4 of this device by thrust/rod is connected with distribution valve 3 of second step/stage of intensification, governing piston 2 of hydraulic performing actuating cylinder, connected with load (aircraft control).

During the deviation of aircraft from the predetermined course sensing element of autopilot, which reacts to this deviation, displaces jet pipe 1, as a result occurs the redistribution of the pressure of the compressed air between the receiving windows a and the cavities of pneumatic actuator 6, which causes the appropriate deformation of diaphragm/membrane 4 and displacement through the thrust/rod of 5 valve 3 of hydraulic part of the drive. Since the valve "follows" the piston of hydraulic cylinder aircraft control surface will be displaced toward the required side, removing the deviation of aircraft from course.

Pneumatic systems of the automation of machining operations.

Especially widely pressure-operated devices are applied for the automation of operations in the machine tools: loading and the attachment billets; the start and the disconnection of the working

class movements of supports; the release and the removal/distance of billets from machine tool; the intra- and between-mill transport of billets; braking operating units with cessation; the supply and the diversion/tap of detents and etc. Furthermore, the pressure-operated devices in the systems of numerical programmed machine-tool control read program. Page 312.

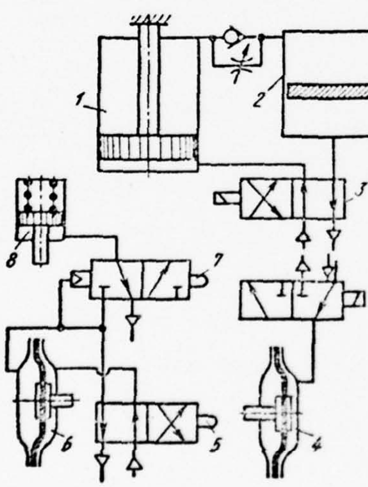

Fig. 257.

Fig. 257. Diagram of pneumatic system with azimuth guidance.

Figure 257 depicts the schematic of the automated lathe with the electropneumohydraulic system of azimuth guidance. Diagram has two diaphragm actuating mechanisms, one of which 6 serves for the drive of the mechanism of the terminal of billet, and the second 4 serves for the start of the clutch of longitudinal feed. Pneumatic cylinder 8 serves for braking arbor, and pneumatic cylinder 1 with tank 2 it serves for the feed of cross-slide. Billet is establish/installed to machine tool by hand, whereupon is switched on auxiliary electric motor for the rapid adjustments of support. The latter affects to distribute 5, establish/installing it in the position of the feed of the compressed air in actuating mechanism 6, that fixes billet, and simultaneously through distributor 7 into pneumatic cylinder 8 whose stock/rod, moving upward, free/releases the brake of arbor. With brake is connected electric-terminal disconnection switch, which at the end of the releasing the brake involves the motion of longitudinal feed. Begins the process of machining.

At the end of this operation is switched on the electromagnet of distributor 3, through which the compressed air enters the pneumatic cylinder of 1 support of cross feed whose stock/rod is rigidly connected with the housing of machine tool. At the end of the course of support is switched on the electric motor for the out-rapid

traverse of the turret saddle. In this case the air is supplied to tank 2, displacing from it oil into the hydraulic cavity of pneumatic cylinder 1. At the end of the back stroke of the turret saddle the electric motors are disconnect/turned off, arbor brakes, and billet is free/released.

Pneumatic reader.

Figure 258 shows the schematic of pressure-operated device for the reading of the program, plotted/applied on motion-picture film in the form of the combination of opening/apertures.

Air under pressure 3-4 kgf/cm² comes from grid/network channels a and b of slide valve box 4, in which is located plunger 13 and given by electromagnet 7 slide valve 5. Upon the switching on of electromagnet 7 valve 5, compressing spring 6, is moved to the left position, in which the compressed air enters through the opening/aperture of valve from channel b under plunger 13 and moves it upward. In this case rubber blanket 8 is dilate/extended and, compressing lath 9, it presses it and the motion-picture film, broached by rachet 14, to reading head 12. During the lift of plunger

13 is open/disclosed also channel c, through which the air enters channel d of reading head and into uptakes e and f ($\varnothing$ = 1.2 mm) and by the further directed jets to film/strip. At the current torque/moment in motion-picture film is a opening/aperture against jet, air it passes through it, but if jet is overlapped by film/strip, then air it passes into the lateral sloping channels and, acting on contact plates, it closes contacts 10 and 11, as a result appears electrical signal. With the aid of these signals occurs the reading of the written on tape program. Page 312.

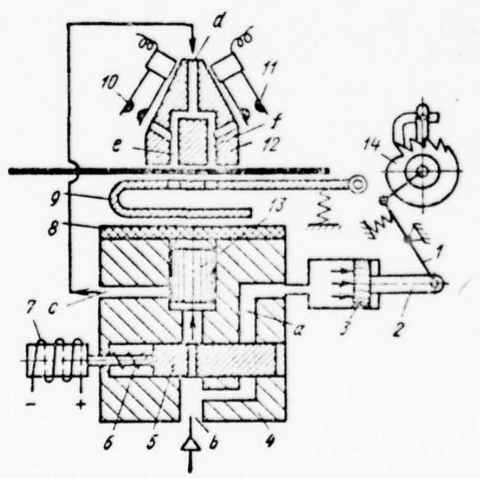

Fig. 258.

Fig. 258. Schematic of pressure-operated device for the reading of program. / ¶ Simultaneously with this pusher 2 under the spring effect of lever 1 is moved to the left and sets the rachet of 14 tape-drive mechanisms.

With the disconnection of electromagnet 7 valve 5 under spring effect 6 is moved to the right position, detaching channel b from channel under plunger 13. As a result plunger 13 is omitted and free/releases motion-picture film, the compressed air it passes from channel a to the piston of 3 pushers 2 and moves it to the right to detent. In this case lever 1 through another lever and the detent turns rachet 14, displacing tape to the value of the selected space.

Instruments of the systems of pneumatic automatics with the diaphragm/membranes of variable effective area.

In the systems of pneumatic automatics are common the instruments, constructed on the basis of membrane/diaphragm node/unit with variable effective area (see Fig. 229). Figure 259a depicts the amplifier circuit of the pressure, which is applied in the different pneumatic systems of monitoring and industrial control. Amplifier

makes it possible to ensure the operations, described by the equality

$$p = kp_1,$$

where p and p_1 - air pressure at output/yield from device and at the inlet into it; k is the transfer function of device, which in the majority of cases is constant factor.

Amplifier consists of the command two-membrane/diaphragm of chamber a with variable effective area and converter of the type" nozzle" - shutter/valve 2-3, switching on the two-membrane/diaphragm node/unit b with negative feedback. Converter nozzle - shutter/valve includes nozzle 2, the throttle/choke of fixed resistor 4 and shutter/valve 3.

The command chamber a is formed by two diaphragm/membranes with adjustable effective areas of S_1 and s_1, but chamber b of negative feedback is formed by diaphragm/membranes with areas of S_2 and s_2. All the indicated diaphragm/membranes are united into common/general/total rigid block (center) 1.

Usually the diameters of the housings of rigid centers and

washers of the jamming of small (s_1 and s_2) and large (S_1 and S_2) diaphragm/membranes are respectively equal to each other.

Factor of amplification (transfer function) k is regulated by displacement with the aid of the screw/propeller of 5 shutter/valve 3, which follows nozzle 2, connected with membrane/diaphragm block 1. In moving membrane/diaphragm block 1 occurs a simultaneous change in the effective areas of all diaphragm/membranes and a change in the coefficient, amplifying device.

Equilibrium condition of membrane/diaphragm block 1 under the static behavior

$$p_1 (S_1 - s_1) = p (S_2 - s_2),$$

whence the controlled pressure p at the output/yield of the instrument

$$p = p_1 \frac{S_1 - s_1}{S_2 - s_2} = kp_1,$$

where

$$k = \frac{S_1 - s_1}{S_2 - s_2}.$$

Page 314.

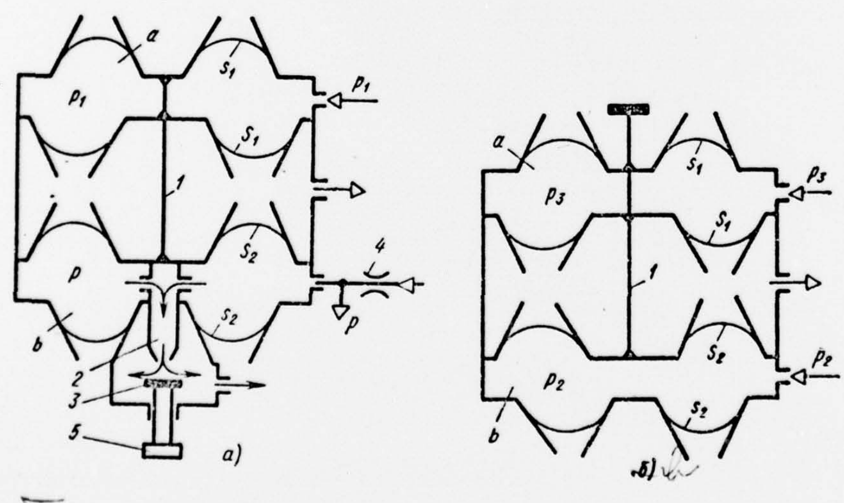

Fig. 259.

Fig. 259. Pneumos-instrument with the diaphragm/membranes of variable area. ¶ Since the effective areas of each of two bellows are determined by differences in the effective areas of the corresponding paired diaphragm/membranes (see page 284), device in question it has the wide (virtually not limited) range of intensification $(0 \leqslant k \leqslant \infty)$.

Angle of taper of the rigid centers of diaphragm/membranes and washers of jamming usually about 30°. In devices of this type usually is observed for obtaining the maximum range of a change in the amplification factor the condition of the equality of the maximum effective area of a small diaphragm/membrane s_{max} and of the minimum effective area of large diaphragm/membrane S_{min}:

$$s_{max} = S_{min}.$$

Diaphragm/membranes are manufactured predominantly from rubberized twill fabric 0.2-0.5 mm thickness.

Experiments show that an error in the amplifier of such type

(diameters of the washers of the jamming of the large and small diaphragm/membranes of the tested instruments are respectively equal to 42 and to 30 mm, but the diameters of the apex/vertexes of the truncated cones of the rigid centers of small and large diaphragm/membranes - 9 and 19 mm) during a change of the input and output signals of pressure in the range 0.2-1 kgf/cm^2 does not exceed ± 0.5o/o.

Figure 259b shows the schematic of the pneumatic position device, intended for the transformation of the pneumatic analog signals of pressure p_2 and p_3 into the displacement/movement of output/yield (for example, the shutter/valve of an amplifier of the type" nozzle" - shutter/valve). This device includes two diaphragm chambers a and b, from which chamber a is formed by small and large diaphragm/membranes with adjustable effective areas of s_1 and S_1, but chamber b - by diaphragm/membranes with areas of s_2 and S_2.

The rigid centers of all diaphragm/membranes are united into common/general/total block 1, in moving which change the effective areas of all diaphragm/membranes.

The angles of taper of washers and rigid centers of all diaphragm/membranes are usually identical (approximately 30°). Furthermore, the areas of small (s_1 and s_2) and large (S_1 and S_2) diaphragm/membranes are respectively equal. Page 315.

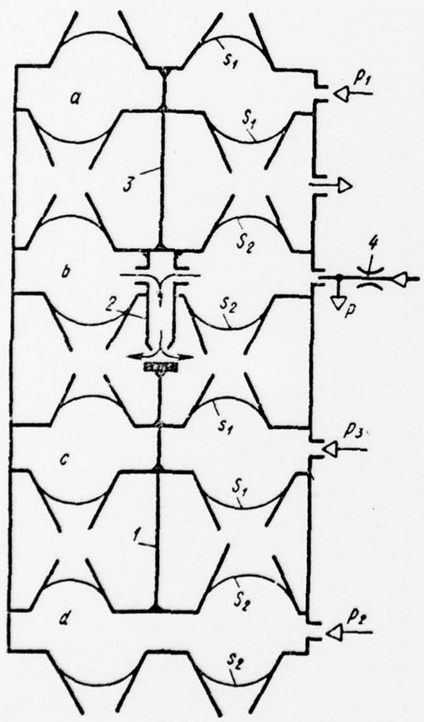

Fig. 260.

Fig. 260. Pneumatic analog computer. Besides this, is observed the condition $S_{max} = S_{min}$, by which will be provided the maximum displacement/movement of block 1 depending on signals p_2 and p_3.

the condition of the static equilibrium of block 1 under effect of pressure in chambers a and b

$$p_3 (S_1 - s_1) = p_2 (S_2 - s_2),$$

whence we find

$$\frac{p_3}{p_2} = \frac{S_1 - s_1}{S_2 - s_2}.$$

A change in the left side of the equality (relation of signals p_3/p_2) produces change in the right side and a change in the position of the membrane/diaphragm unit of position device.

The experiment showed, that an instrument error of the same size/dimensions, as in the case examined above, during a change of the input signals p_2 and p_3 in the range 0.2-1 kgf/cm² does not exceed 2.5o/o.

On the basis of the described membrane/diaphragm amplifiers of pressure and position device can be constructed pneumatic analog computer for multiplication and division of two independent variables, and also for squaring and root-squaring of one variable. the amplifier of pressure with diaphragm chambers a and b (Fig. 260, see also Fig. 259a) and position device with chambers c and d (see also Fig. 259b) are connected in such a way that the rigid center of position device is the shutter/valve of nozzle 2, connected with the stock/rod of 3 amplifiers of pressure. Since the end/face of the stock/rod of 1 position device serves as shutter/valve for the nozzle of 2 amplifiers of pressure, nozzle follows the displacement/movements of stock/rod 1, as a result are provided the equal displacement/movements of stock/rods 3 and 1. Air under pressure p will be fed into chamber b of the feedback of the amplifier through throttle/choke 4 of fixed resistor. The signal of pressure p, form/shaped in chamber b of feedback, is the result of the performed by this device mathematical operation. To the inlet of amplifier into chamber a is supplied signal p_1, also, to chambers c

and d of position device - signals p_2 and p_3. Chambers a of the amplifier of pressure and c - position device are formed by identical diaphragm/membranes with areas of s_1 and S_1, but chamber b and d - by diaphragm/membranes with identical areas of s_2 and S_2.

From the analysis of the interaction of amplifier and position device, produced taking into account the equality of the areas of the diaphragm/membranes, which form chambers a and c, and also b and d, and series communication/connection of stock/rods 3 and 1 (*computational operations are not given*), it follows that the work of the computer in question is described by the equation

$$p = \frac{p_1 p_2}{p_3}.$$

Page 316.

[redacted]/appendix. Relationships between the units of the physical quantities.

Наименование величины	Единица: Наименование	Единица: Обозначение	Значение в единицах СИ, кратных и дольных от них
Масса	Килограмм-сила-секунда в квадрате на метр	$кГ \cdot сек^2/м$	9,80665 $кг \approx 10$ $кг$
Плотность	Килограмм-сила-секунда в квадрате на метр в четвертой степени	$кГ \cdot сек^2/м^4$	9,80665 $кг/м^3 \approx 10$ $кг/м^3$
	Килограмм на кубический сантиметр	$кг/см^3$	10^6 $кг/м^3$
Сила, вес	Грамм на кубический сантиметр	$г/см^3$	10^3 $кг/м^3$
	Килограмм-сила	$кГ$	9,80665$Н \approx 10Н$
	Грамм-сила	$Г$	9,80665 $мН$
	Тонна-сила	$Т$	9,80665 $кН$
	Дина	$дин$	10^{-5} $Н$
Момент силы	Килограмм-сила-метр	$кГ \cdot м$	9,80665 $Н \cdot м \approx 10$ $Н \cdot м$
	Килограмм-сила-сантиметр	$кГ \cdot см$	98,0665 $мН \cdot м =$ $= 9{,}80665 \cdot 10^{-2}$ $Н \cdot м$
Давление, механическое напряжение	Килограмм-сила на квадратный метр	$кГ/м^2$	9,80665 $Па \approx 10$ $Па$
	Килограмм-сила на квадратный сантиметр	$кГ/см^2$	$98{,}0665 \cdot 10^3$ $Па \approx$ $\approx 0{,}1$ $МПа$
	Килограмм-сила на квадратный миллиметр	$кГ/мм^2$	$9{,}80665 \cdot 10^6$ $Па =$ $= 9{,}80665$ $МПа \approx$ ≈ 10 $МПа$
	Миллиметр ртутного столба	$мм$ $рт.$ $ст.$	133,322 $Па$
Работа, энергия	Килограмм-сила-метр	$кГ \cdot м$	9,80665 $Дж \approx 10$ $Дж$
Количество теплоты	Калория	$кал$	4,1868 $Дж$
	Килокалория	$ккал$	$4{,}1868 \cdot 10^3$ $Дж =$ $= 4{,}1868$ $кДж$
Мощность	Килограмм-сила-метр в секунду	$кГ \cdot м/сек$	9,80665 $Вт \approx 10$ $Вт$
	Лошадиная сила	$л.$ $с.$	735,499 $Вт$
Динамическая вязкость	Пуаз	$пз$	0,1 $Па \cdot сек$
	Сантипуаз	$спз$	10^{-3} $Па \cdot сек = 1$ $мПа \cdot сек$
	Килограмм-сила-секунда на квадратный метр	$кГ \cdot сек/м^2$	9,80665 $Па \cdot сек \approx$ ≈ 10 $Па \cdot сек$
Кинематическая вязкость	Стокс	$ст$	10^{-4} $м^2/сек$
	Сантистокс	$сст$	10^{-6} $м^2/сек = 1$ $мм^2/сек$
Удельное количество теплоты	Калория на грамм Килокалория на килограмм	$кал/г$ $ккал/кг$	$4{,}1868 \cdot 10^3$ $Дж/кг =$ $= 4{,}1868$ $кДж/кг$
Удельная теплоемкость	Калория на граммградус Цельсия	$кал/(г \cdot {}^\circ C)$	$4{,}1868 \cdot 10^3$ $Дж/(кг \cdot K)$
	Килокалория на килограммградус Цельсия	$ккал/(кг \cdot {}^\circ C)$	
Тепловой поток	Калория в секунду	$кал/сек$	4,1868 $Вт$
	Килокалория в час	$ккал/ч$	1,163 $Вт$
Теплопроводность	Калория в секунду на сантиметр-градус Цельсия	$кал/(сек \cdot см \cdot {}^\circ C)$	418,68 $Вт/(м \cdot K)$
	Килокалория в час на метр-градус Цельсия	$ккал/(ч \cdot м \cdot {}^\circ C)$	1,163 $Вт/(м \cdot K)$

Key: (1). Designation of value. (2). Unit. (3). Value in the units of SI, multiple and lobate from them. (4). Designation. (5). Designation. (6). Mass. (7). kilogram-force-second squared to meter. (8). kg•s²/m. (9). kg. (10). Density. (11). kilogram-force-second squared to meter to the fourth degree. (12). kgf•s²/m⁴. (13). kg/m³. (14). Kilogram to cubic centimeter. (15). kg/cm³. (16). Gram to cubic centimeter. (17). g/cm³. (18). Kilogram-force. (19). Force, weight. (20). gram-force. (21). kgf. (21a). N. (22). G. (23). mN. (24). Ton-force. (25). T. (26). kN. (27). Dyne. (28). dynes. (29). Moment of force. (30). Kilogram-meter. (31). kg•m. (32). N•m. (33). kilogram-force-centimeter. (34). kg•cm. (35). mN•m. (36). Kilogram-force to square meter. (37). kgf/m². (38). Pa. (39). Pressure, mechanical stress. (40). Kilogram-force to square centimeter. (41). kgf/cm². (42). MPa. (43). Kilogram-force to square millimeter. (44). kg/mm². (45). Pa MPa. (46). Millimeter of mercury. (47). mm Hg. (48). Pa. (49). Work, energy. (50). Kilogram-meter. (51). kg•m. (52). J. (53). Calories. (54). cal. (55). J. (56). Amount of heat. (57). Kilocalories. (58). kcal. (59). J kJ. (60). Power. (61). Kilogram-meter per second. (62). kg•m/s. (63). W. (64). Horsepower. (65). hp. (66). W. (67). Dynamic viscosity. (68). Poise. (69). poise. (70). Pa•s. (71). Centipoise. (72). cp. (73). Pa•s mPa•s. (74). kilogram-force-second to square meter. (75). kg•s/m². (76). Pa•s. (77). Kinematic viscosity. (78). Stokes. (79). steel. (80). m²/s. (81). Centistoke. (82). c. st. (83). mm²/s. (84).

Specific amount of heat. (85). Calorie on Gram. (85a). Kilocalorie to kilogram. (86). cal/g. (86a). kcal/kg. (87). J/kg. (88). Specific heat. (89). Calorie to the gram/degree of Celsius. (90). cal/ (g• C). (91). Kilocalorie to kg/degree of Celsius. (92). kcal/ (kg•°C). (93). J/ (kg•K). (94). Heat flux. (95). Calorie per second. (96). cal/s. (97). W. (98). Kilocalorie in hour. (99). kcal/h. (100). W. (101). Thermal conductivity. (102). Calorie per second to the centimeter-degree of Celsius. (103). cal/ (s•cm•°C). (104). W (m•K). (105). (105). Kilocalorie in hour to the meter•degree of Celsius. (106). kcal/ (h•m•°C. (107). W (m•K). Page 317.

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